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The design and development of a sensitive torque and thrust dynamometer for small ship models

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THE DESIGN AND DEVELOPMENT OF A SENSITIVE TORQUE AND THRUST DYNAMOMETER FOR SMALL SHIP MODELS

James Augustine Bortner
and
Benedict Louis Stabile

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DYN - DYNAMOMETER

SHI - SHIP MODELS

Letter on cover:

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SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS

FOR THE DEGREE OF

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at the

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June 1956

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April 13, 1956

Prof. Laurens Troost
Naval Architecture

Dear Prof. Troost:

This is to authorize the deposit in the library of multilithed copies of theses accepted by the Department of Naval Architecture and Marine Engineering from Course XIII-A candidates for the degree of Naval Engineer in lieu of the ribbon copies normally required.

ABSTRACT

THE DESIGN AND DEVELOPMENT OF A SENSITIVE TORQUE AND THRUST DYNAMOMETER FOR SMALL SHIP MODELS

Lt. James A. Bortner, USN

Lt. Benedict L. Stabile, USCG

Submitted to the Department of Naval Architecture and Marine Engineering on 21 May 1956 in partial fulfillment of the requirements for the degreee of Naval Engineer

The object of this thesis was to determine the feasibility of constructing a sensitive torque and thrust dynamometer for use in self-propulsion tests of small ship models. Since the losses experienced in stern tube bearings are of the same order of magnitude as the quantities to be measured in these small models, it was proposed to accomplish the measurement between the propeller and the point of entry of the propeller shaft into the hull.

Early experimentation centered around the use of strain gages mounted on a plastic material with low modulus of elasticity. A satisfactory prototype of a torque measuring gage was constructed from these materials. Failure to develop a suitable thrust measuring gage from these materials led to the construction of a thrust gage based on the proving ring principle, and another based on the use of a linear variable differential transformer.

Static calibration curves of each of these devices are given and the shortcomings, as well as the good points of each is discussed. Mention is made of the difficulties encountered due to slip-ring noise when dynamic measurements were attempted. The final choice of the differential transformer thrust gage in preference to the ring type gage was made to gain a greater output signal in the face of these noise problems.

The conclusion is presented that the prototype design constructed by the authors is not satisfactory. The authors state that they feel that the proposed installation is entirely feasible and that their prototype design offers great potential for future development. A number of recommendations are made dealing with design improvements, construction methods, and testing procedures which it is felt will lead to a wholly acceptable dynamometer.

Thesis Supervisor: Martin A. Abkowitz

Title: Associate Professor of Naval Architecture

Cambridge, Mass.
21 May, 1956

Professor Leicester F. Hamilton
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge 39, Massachusetts

Dear Sir:

In accordance with the requirements for the Degree of Naval Engineer, we submit herewith a thesis entitled: "The Design and Development of a Sensitive Torque and Thrust Dynamometer for Small Ship Models".

ACKNOWLEDGMENT

The authors wish to express their appreciation for the guidance and encouragment given them by Professor M. A Abkowitz in carrying out this Thesis. In particular, Professor Abkowitz' suggestion of the possibility of utilizing a Linear Variable Differential Transformer in the thrust measuring device is appreciated as having contributed in large measure towards the development of a useful dynamometer prototype.

Special acknowledgment is made for the interest, technical advice and equipment provided by Professor W. M. Murray during the course of this investigation.

Much gratitude is due Professor J. F. Reintjes for his aid in enlisting the services of the M. I. T. Servomechanisms Laboratory, at which establishment the Differential Transformer thrust gage was fabricated.

Finally, acknowledgment is made to Mr. R. E. Johnson of the M. I. T. Propeller Tunnel and Mr. S. N. Blake of the M. I. T. Ship Model Shop for the valuable assistance in the various construction phases of this Thesis.

James A. Bortner
Benedict L. Stabile

Cambridge, Massachusetts
May 21, 1956

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I. INTRODUCTION

Estimates of ship resistance are quite commonly made by towing small ship models in small tanks.* Self-propulsion tests are, however, generally carried out on larger (20 to 30 foot) models. The use of this size models requires large towing tank facilities such as the U.S. Navy's David Taylor Model Basin and expensive equipment.

The object of this thesis is to investigate the problems inherent in the design of a suitable torque and thrust measuring device for small ship models, and to develop such a design if it is considered feasible. The successful production of such a device would allow self-propulsion tests to be made in small tanks with a consequent reduction in the expense of testing. One factor which has probably retarded the previous development of such equipment was a feeling that scale effects would be so great that no reliance could be placed upon the data obtained.

At least one effort (16) has been made to conduct self-propulsion tests in a small tank by towing standard five foot models for resistance data and then conducting self-propulsion tests on 10 foot models. The use of two models (aside from the obvious disadvantage of requiring a second model) calls for correlation of data from two tests and correction for wall effects when using the oversize model. The need for such data correlation and correction is eliminated if both resistance and self-propulsion tests are carried out on the 5 foot model only. The difficulties encountered when one attempts to carry out the self-propulsion tests on the small models are:

* A small towing tank is generally considered to be one of 10 feet or less breadth and less than 150 feet long. The small models towed in these tanks are approximately 5 feet in length.

[illegible]

1. The device must be physically small to fit inside the model and light enough that the model will not exceed design draft.

2. The torque and thrust developed by the propeller are very small, being on the order of one inch-ounce of torque and one-fourth pound of thrust at full speed.

The quantities to be measured are of the same order of magnitude as the losses one would expect in the stern tube bearing of the model. Thus one concludes that the measurement should be made between the propeller and the point of entry of the shaft into the hull. This conclusion leads to the necessity for some length of overhanging propeller shaft to accomodate the measuring device. Such an overhang may lead to whirling vibrations which are undesirable because they are likely to destroy hydrodynamic similarity between model and full scale ship. In addition, these vibrations will necessitate making the measuring device unresponsive to bending of the propeller shaft.

There are, in principal, many devices which will measure torque and thrust independently of bending and also independently of each other. Some of these useful devices are:

1. Electrical resistance strain gages
2. Electrical inductance gages
3. Electrical capacitance gages
4. Piezo-electric devices
5. Optical devices
6. Micro-wave cavity resonators
7. Vacuum tube gages

These devices are described in their various forms in reference (9) and in greater detail in (2), (3), (4), (7) and (10).

Common to all the devices mentioned above is the fact that a deflection or force must generate an electrical signal that will have a signal to noise ratio great enough for accurate measurement and calibration. In most applications these deflections and forces are large thus permitting the use of high modulus materials which are generally associated with low creep and low hysteresis losses. A few simple calculations, however, (see Detailed Procedure, Appendix A) show that for the order of torque and thrust involved in this design materials of low modulus, such as plastics, may have to be utilized. An alternative solution is to use materials of high modulus with cross section sufficiently reduced to produce detectable signal outputs to the sensing device.

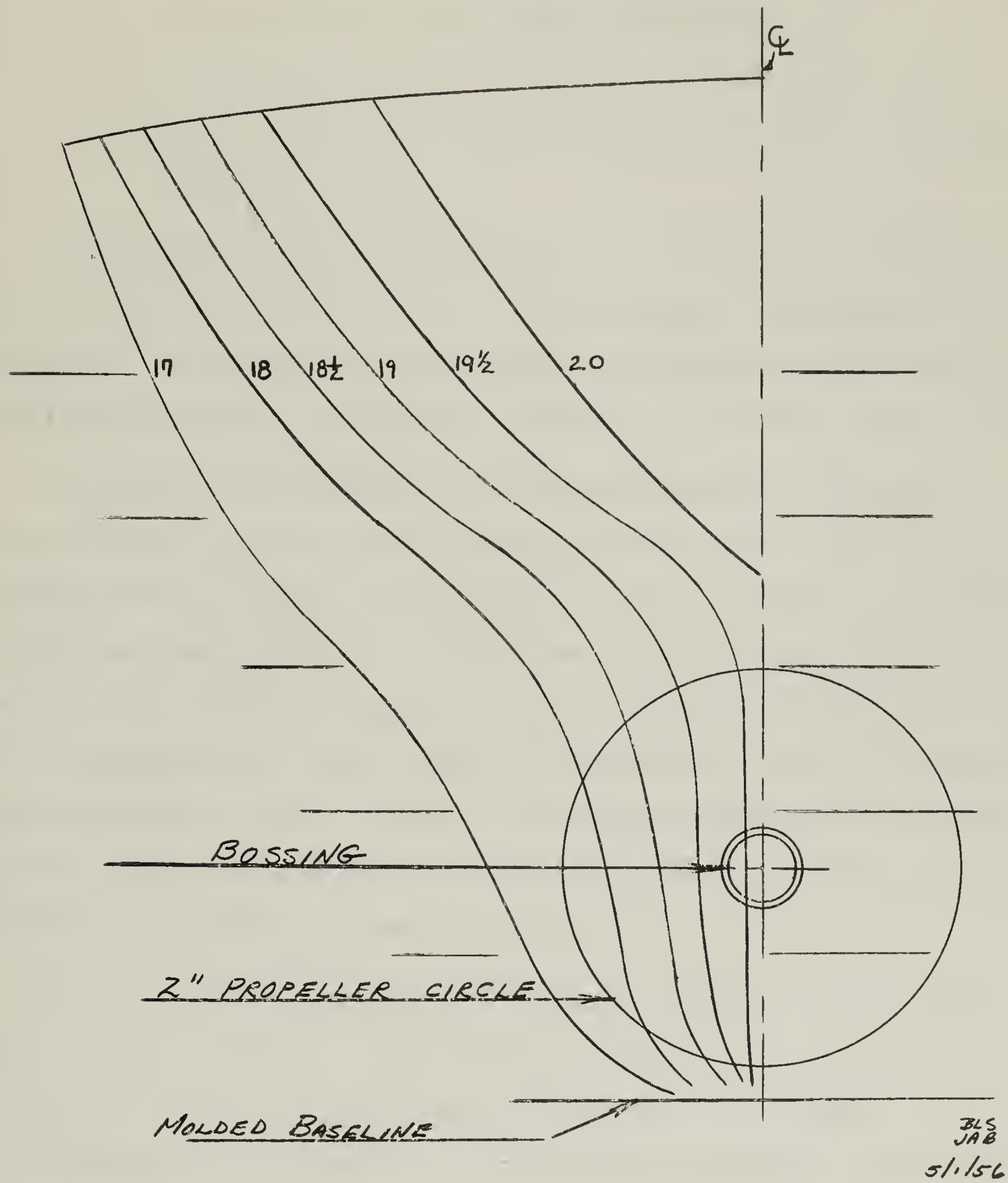
The devices mentioned previously have been successfully utilized in the aircraft, structural, and automotive fields for measurement of torque and thrust. Measurements have been made on high speed rotating shafts such as turbine shafts and transmission shafts and the signals have been removed by specially designed slip ring arrangements.

A survey of the literature of the past twenty years indicates an ever increasing successful use of gages for the measurement of torque and thrust on rotating shafts. No examples of an installation in which the signals to be detected were of such a small order of magnitude as is proposed in this thesis were found. There were, however, no basic objections found which would indicate that the project could not be carried out.

Since it was desired to have the dynamometer adaptable for use in any five foot ship model, the Series 60, Block 0.60 model was chosen as the model for which the dynamometer would be designed. A partial body plan which illustrates the fine lines of the stern sectinn of this model is shown in Figure I.

FIGURE I

PARTIAL BODY PLAN SHOWING STERN SECTIONS OF SERIES 60 BLOCK 0.60 SHIP MODEL



SCALE: FULL SIZE

II. PROCEDURE

Resistance strain gages were chosen from the devices mentioned previously as the sensitive elements to convert the mechanical strains generated by torque and thrust into electrical signals. This choice was based on the following factors:

(a) The cost of these gages is moderate.

(b) They are available in sizes as small as 1/16 inch.

(c) A relatively high degree of precision can be obtained using these gages.

In addition to the factors just mentioned, the theory covering strain gage installations for purposes such as proposed by this thesis is well established. References (1) and (9) discuss installations made on larger scale applications than we propose; the theory, however, is the same.

Preliminary investigations, described in detail in Appendix A, led to the selection of a plastic as the shaft material. The choice fell to Vinylidene Chloride (Saran) on the basis of its low modulus of elasticity, excellent resistance to water and solvents, and good machinability. A combined torque and thrust gage was fabricated by machining a Saran rod and mounting two four-gage bridges. Preliminary static tests of this device disclosed several serious defects. The worst of these was a high creep rate. For this reason, tests of this device were discontinued and attention was focused on the selection of another plastic.

It was felt that although smaller strains would be obtained from a plastic of higher modulus, the elimination of the high creep rate experienced with Saran would more than compensate for the loss in signal. Accordingly, new torque and thrust shafts were constructed by mounting strain gage bridges on Lucite tubing.

Although static tests of the torque shaft were encouraging, the thrust shaft tests indicated excessive response to bending and low response to

thrust. These difficulties seemed to stem from inaccurate alignment of the strain gages and excessive cross-sectional area of the shaft. Several attempts to construct new thrust gages with smaller cross-sectional area failed due to splitting of the plastic when the wall thickness was reduced the necessary amount. As a result of these continued problems with the plastic thrust gage it was decided to continue with the Lucite torque shaft but to investigate different types of thrust measuring gages.

The next attempt to produce a suitable thrust gage was based on the proving ring principle. A thin brass ring was inserted in the propeller shaft with its axis perpendicular to the shaft axis. Deflections of this ring under load would produce strains which could be measured by a strain gage bridge mounted on the ring. Although the static calibration of this gage showed a linear response, the magnitude of the response was small. In addition, the assembly was not sturdy enough to prevent an unacceptable amount of whipping in the propeller shaft.

Concurrently with the experiments on the ring gage another thrust gage was constructed utilizing a Schaevitz linear variable differential transformer (22) as the sensing element. The transformer coil was rigidly mounted in a sleeve. The transformer core was threaded and inserted in the propeller shaft. This shaft assembly was then supported co-axially with the sleeve in which the coil was mounted by means of a double spring suspension. The spring suspension was designed to allow the transformer to develop full output when the propeller shaft and core moved axially over a distance of 0.010 inch. The whole assembly is designed to rotate with torque being transmitted from the propeller shaft to the outer sleeve by the spring suspension. The outer sleeve was made to be inserted in the hollow end of the Lucite torque shaft. The differential transformer thrust gage is illustrated in Figures IX and VIII-A

Static tests of the differential transformer thrust gage appeared promising. A number of calibration runs were made in order to try to establish the reproducibility that could be anticipated. The results of

these tests will be discussed in sections III.

In order to make dynamic tests, a test stand consisting of a drive motor, slip-ring assembly, and tachometer was constructed. The drive motor was a shunt wound D. C. motor designed to develop 4 inch-ounces of torque at 5400 rpm at rated voltage (27 V D C). The motor shaft extended from both ends of the motor housing. The AC tachometer for speed measurement was coupled to the forward end. The slip-ring assembly for removing signals from the rotating shaft was connected to the after end of the assembly. The construction of the slip-ring assembly is discussed in detail in section A-7 of the Appendix. The completed test stand is shown in Figure XVI.

No successful dynamic tests were conducted in the time available due to the high slip-ring noise experienced and due to the extreme amount of whirling experienced when the assembled torque and thrust gage was rotated. Details of these problems will be discussed in Section III.

III. RESULTS AND DISCUSSION

1. Torque Gage

The construction and testing of the(Lucite shaft-strain gage) torque measuring device is described in detail in Appendix A. Tests were made to determine bending sensitivity as well as to calibrate the torque response.

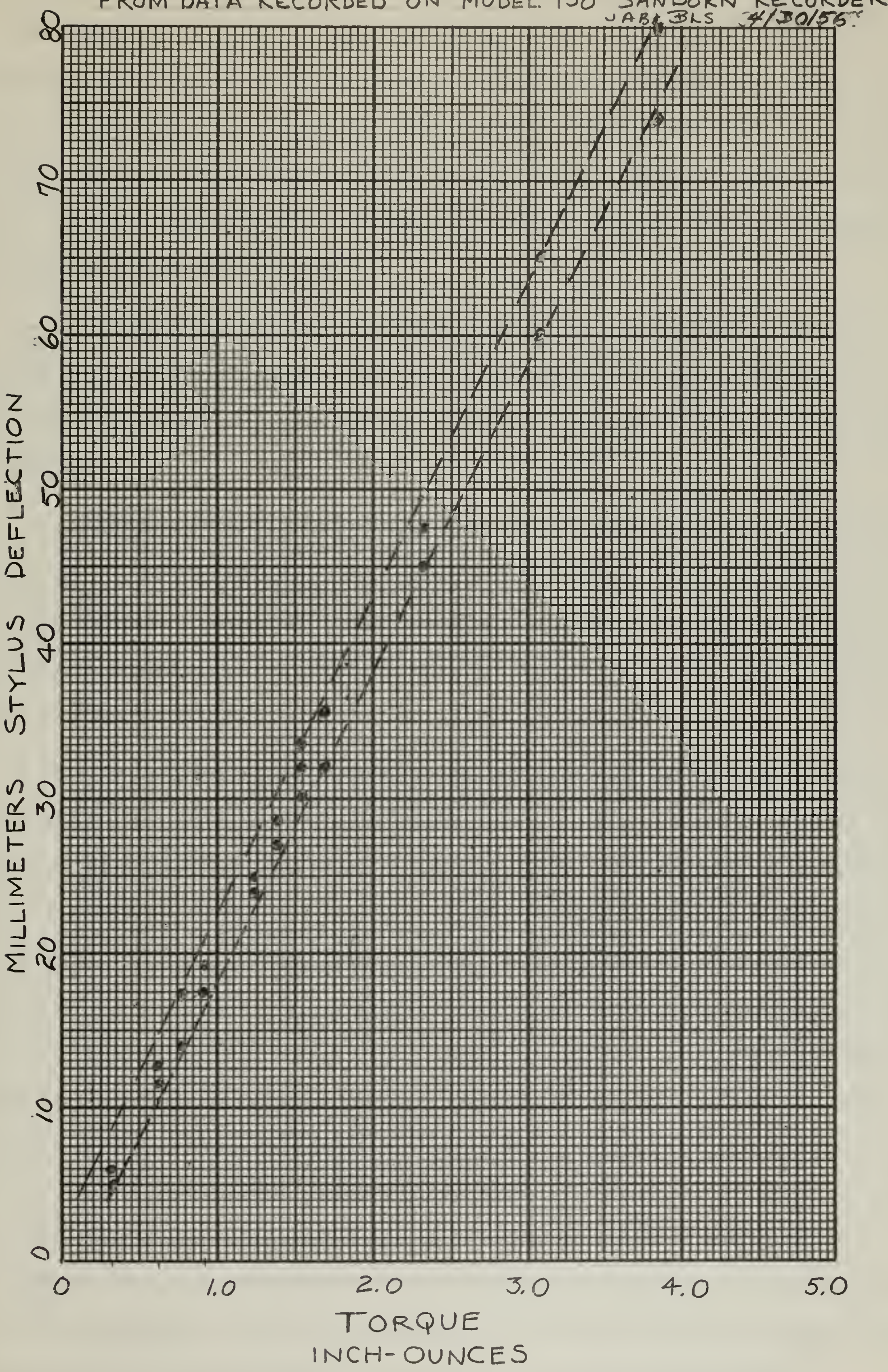
The results of the ten torque calibration runs are plotted in Figure II. The first four runs were made using weights which produced torques in the range from 0.308 to 1.848 inch-ounces. The remaining six runs were made using weights which produced torque increments of 0.77 inch-ounces to a maximum value of 3.85 inch-ounces.

The data plot discloses the fact that, for any given run, the response was linear. The straight lines for successive runs are, however, displaced from one another. This displacement appears as a translation in the stylus deflection (ordinate) of the curves. This is the same as saying that there was a zero shift inasmuch as all lines have equal slopes but do not pass through the same origin. The spread appeared to be independent of the applied torque and was approximately 5 millimeters of stylus deflection at any given torque. With this behavior the error represented by the spread decreases as the torque load is increased. The reason for this behavior is not known but there are several effects which taken singly or in combination could account for it.

The first possibility concerns the explanation of a phenomenon noted during experimentation. When the torque gage was connected to the recorder and sitting idle, with no load applied, a slow drift of the recording stylus was noted. This occurred long after the necessary warm-up period had been completed with all balancing adjustments made. Such a change in stylus position corresponds to a change in the zero setting of the apparatus and could easily account for the observed spread in readings. One likely source of this error is the effect of temperature changes on the bridge. There are two ways in which these temperature effects can make themselves

FIGURE II

STATIC CALIBRATION OF LUCITE TORQUE GAGE
FROM DATA RECORDED ON MODEL 150 SANBORN RECORDER



felt. First, in the event that one gage has a different temperature coefficient of resistance than the remainder of the bridge, an unbalance will result when all gages are exposed to the same temperature change. The second case is that in which all four arms of the bridge have equal temperature coefficients of resistance but do not all experience the same rise in temperature. In the case of gages mounted on a plastic, with relatively low thermal conductivity, it is quite conceivable that random air currents could produce such an effect.

In discussing causes of baseline drift, Reference (25) states that drift due to these causes might be of the order of 2 to 5 millimeters of stylus deflection and the drifts experienced agreed with this value quite closely.

Another possible cause of the shift noted previously would be a hysteresis effect in the plastic. It is considered doubtful that this effect could account for the variations experienced since the variations were both positive and negative in direction whereas the applied torques were always in the same direction.

A final experimental effect noted which could explain the shift concerns the use of the Sanborn recording equipment. It was observed that, immediately after balancing, the stylus would remain quite steady for periods of 20 or more seconds. If, however, the attenuation setting was changed from the X1 scale (where final balance is performed) to another scale, drift immediately commenced. Once this drift commenced it continued even though the attenuator was returned to the X1 scale. It was theorized that the switching action might cause a change in contact resistance, and hence in the overall attenuation, that would account for the base line drift. This theory was rejected since a study of the schematic diagram for the recorder given in Reference (25) shows that the balancing function is isolated from the attenuator. This schematic also shows that the resistances in the attenuator circuit are so large that the probability of contact resistance fluctuations great enough to be detected are extremely unlikely.

As noted earlier, a test was also made to determine the response of the torque gage to bending. This response was found to be negligible. In an effort to obtain quantitative results, the recorder was balanced with the gage factor set at 2.0, attenuation at X1, sensitivity maximum, and zero suppression out. An unrealistically high bending moment of 3.025 inch-pounds was applied to the shaft and a 12 millimeter stylus deflection was observed. In actual running conditinns bending moments are expected to be generated on the propeller shaft due to motion of the model in a seaway and also due to the fact that the center of thrust is below the center of the propeller. Both effects would be small, producing a combined bending moment of only several inch-ounces. It is anticipated that stylus deflections on the order of 10 to 15 millimeters will be developed when the model is operating with 1.5 inch-ounces of driving torque(with recorder set at X5 attenuatinn) on the basis of the calibration obtained. If it is assumed that the combined bending moments due to motion in a seaway and eccentric loading amounts to 5 inch ounces, then one could expect the maximum deflection due to this load to be 0.25 millimeters. This would set an upper limit of 2 1/2% on the torque measurement error as a result of bending moment. The authors feel, however, that there is very little likelihood that the error due to bending would be this great.

In conjunction with the torque calibrations discussed earlier, a series of observations were made for the purpose of computing the transducer gage factor. The procedure for this determination is explained in detail in Reference (25) and will not be reproduced here. As a result of these observations, the gage factor was determined to be approximately 52 micro-volts per volt input per inch-ounce of applied torque. This data is plotted in Figure III. Important points to be noted are the insensitiveness of the gage factor to lead, within the range of loads applied, and the spread of values about the mean value of 52 microvolts per volt per inch-ounce. This spread may be correlated with the spread obtained in the torque calibration runs and is, no doubt, attributable to the same causes.

FIGURE III

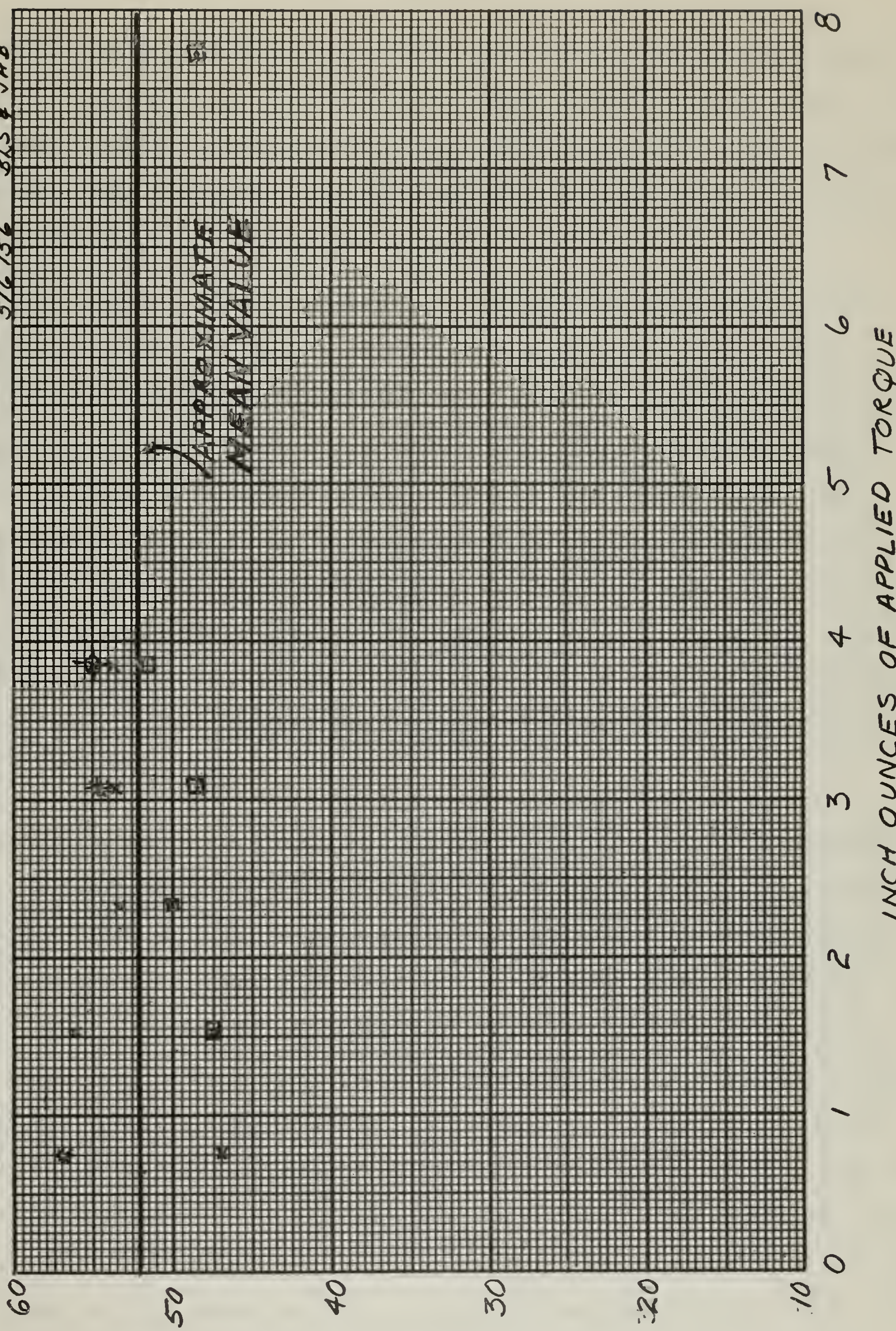


FIGURE III

GAGE FACTOR OF LUCITE TORQUE GAGE - AS COMPUTED FROM ZERO SUPPRESSION MEASUREMENTS WITH VARYING APPLIED TORQUE

2. RING GAGE

The Ring Thrust Gage design and construction is described in detail in section 8 of Appendix A. Following completion of construction, a test was made to determine the response of the gage to thrust and to bending. The ring gage was also connected to a drive shaft and rotated to determine the amount of whirling that might be experienced in use.

The results of the thrust measurement tests are given in the plot reproduced as Figure IV. Examination of the plot reveals that for thrust loads of less than one-quarter pound (0.176 lb. was the maximum load applied) the response of the gage is extremely linear. The output obtained from the gage was, however, of small magnitude. With the attenuator setting, at X1, and the gage factor set at 1.72, the response to the 0.176 pound thrust load was only 30.75 millimeters. Extrapolating to a full load thrust of 0.25 pounds the total deflection would still be less than 40 millimeters. Since this signal would, in the dynamic installation in the ship model, have to be passed through a set of slip-rings, the necessity for extremely low slip-ring noise becomes very apparent.

The ring gage was rotated in the test stand at a speed of approximately 2000 rpm. Severe whirling of the end of the shaft resulted due to run-out in the assembly. The unit was stopped and an attempt was made to straighten the assembly using simple finger pressure. A very marked improvement was noted and it was decided that an attempt at a dynamic calibration should be made. Connecting the gage to the recorder through the slip rings, the standard balancing procedure was followed with the assembly stationary. When the drive motor was started wild oscillations of the stylus were observed and no useful information whatsoever was recorded on the tape. The slip rings were then polished with 6/0 sandpaper and buffed with a piece of canvas. Following this treatment the equipment was again connected and another test made. The slip ring noise had been reduced to a very rapid oscillation of considerably reduced magnitude. Load pulses of approximately a pound and a half were

FIGURE IV

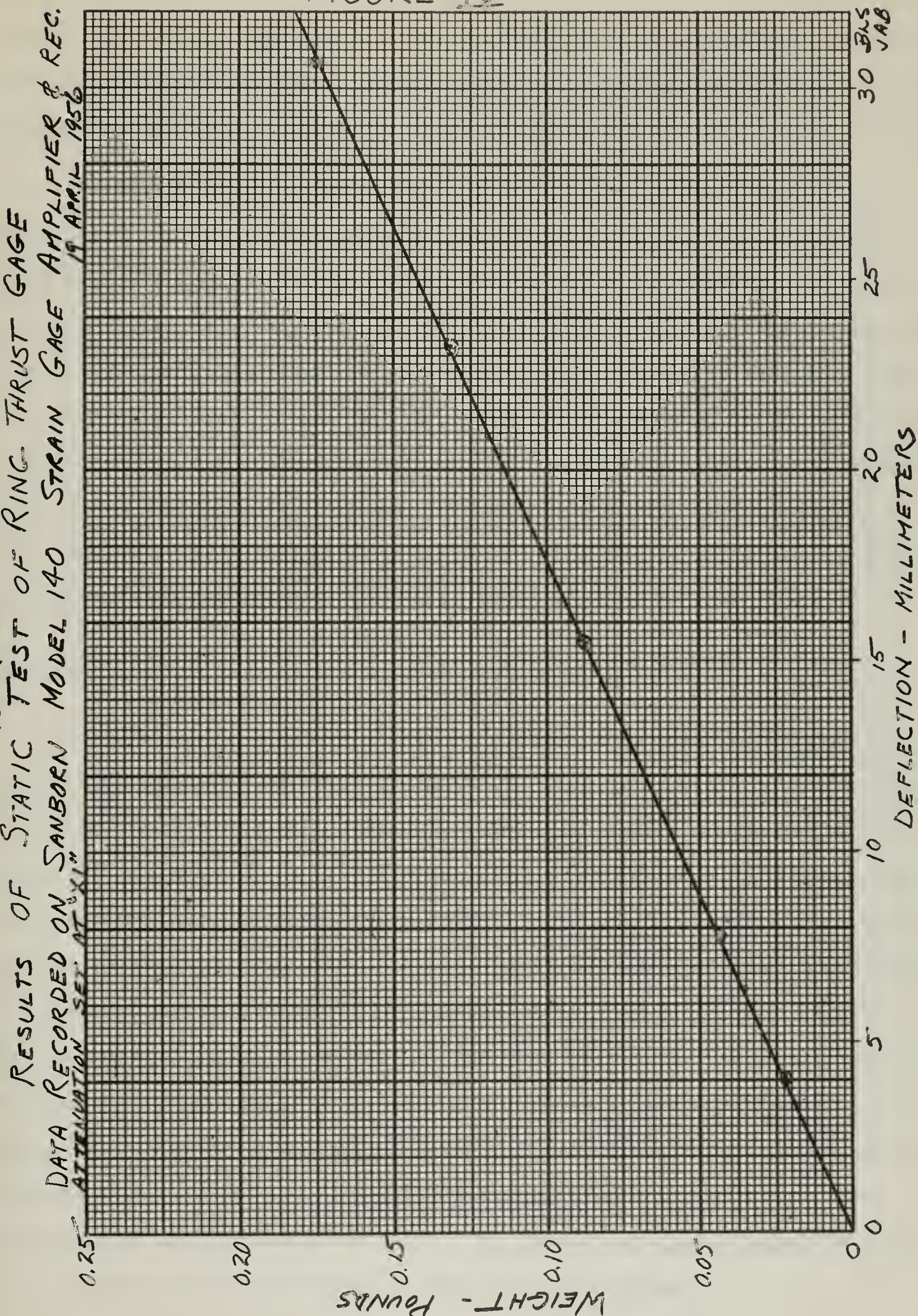


FIGURE IV
RESULTS OF STATIC TEST OF RING THRUST GAGE
DATA RECORDED ON SANBORN MODEL 140 STRAIN GAGE AMPLIFIER & REC.
ATTENUATION SET AT "X1"
19 APRIL 1956
BLS
JAB
DEFLECTION - MILLIMETERS
0
5
10
15
20
25
30
WEIGHT - POUNDS
0
0.05
0.10
0.15
0.20
0.25

applied to the end of the shaft (Attenuator setting had been changed to X20 scale because of the previous large amplitude of oscillations due to slip ring noise). The pulses produced stylus deflections of approximately 15 millimeters. The slip-ring noise was evidenced as a modulation of the load signal with an amplitude very nearly one half as great as that of the desired signal.

In spite of the high noise level, it was decided to attempt a static calibration. As preparations were being made for this effort, the end of the shaft was accidently struck a light blow while it was rotating. The result of this accident was that the extreme whirling vibrations re-occurred and it then became impossible to apply the thrust to the end of the shaft due to its motion. It was decided that this gage was too fragile for the application proposed unless some alterations could be made in the design and further tests were discontinued. Figure XXII included at the end of Appendix A illustrates the reduction of slip ring noise obtained by the slip-ring polishing procedure described.

3. Linear Variable Differential Transformer Thrust Gage

The results of static calibration runs made on the thrust gage are plotted in Figures V and VI. In Figure V the upper and lower lines enclose the points determined in ten separate runs. The data for these curves is contained in Tables IV and V of Appendix B. The table data is recorded from the original recorder paper traces and is included to permit larger scale plots to be made for purposes of study. Figure V shows that the calibration is nonlinear and that there is appreciable spread in the data. Several interesting points may be noted on a large scale plot of the runs. First it was noted that individual runs produced points which formed a fair curve. The points on any one run generally were consistent in that they were either higher or lower than the average curve through the points for all runs. The second point worthy of note is that at the lower and upper ranges of applied load the data spread is smaller than the central region.

At this point it is worth while mentioning some aspects of the gage construction which might have some bearing on these characteristics.

FIGURE V

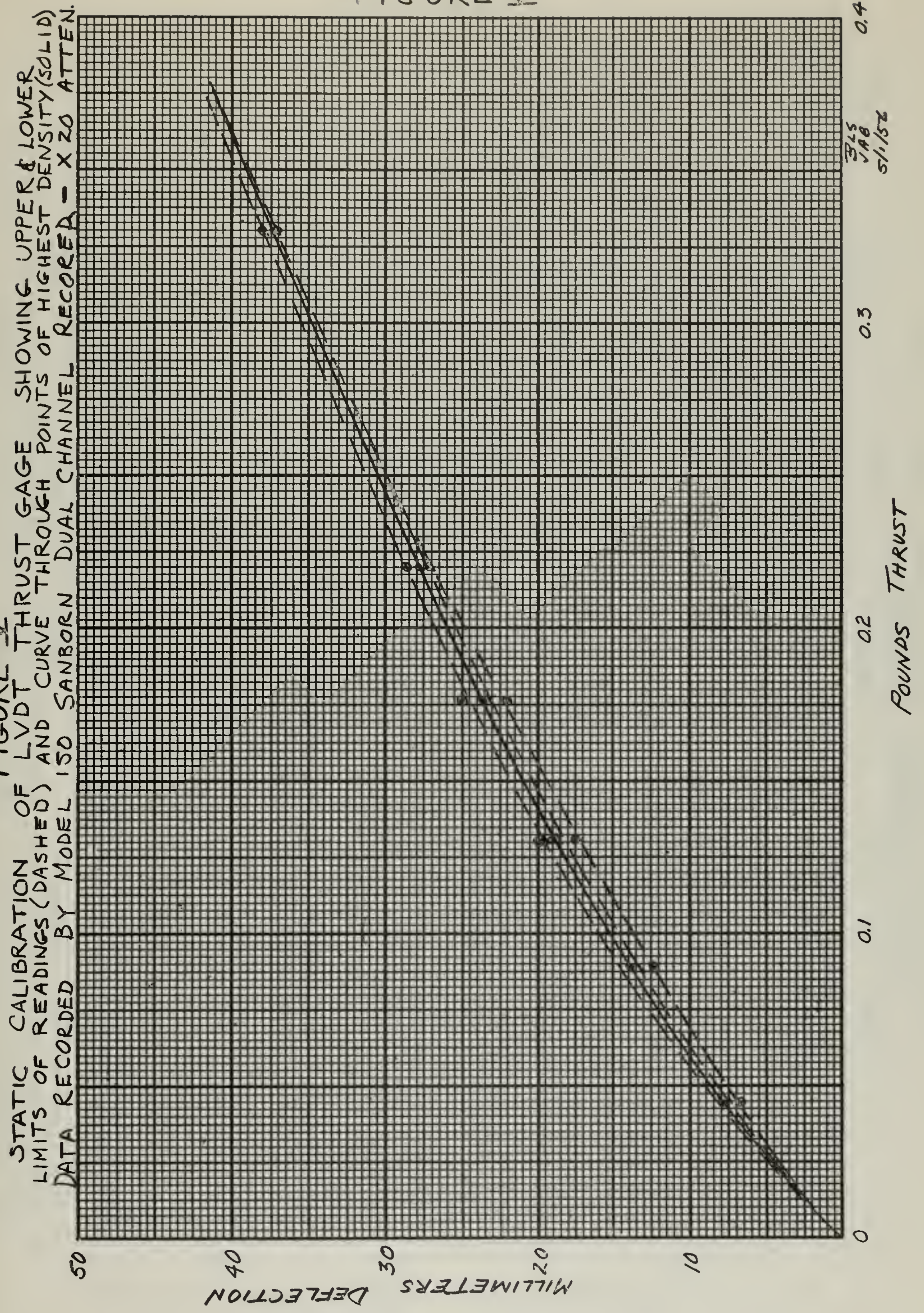
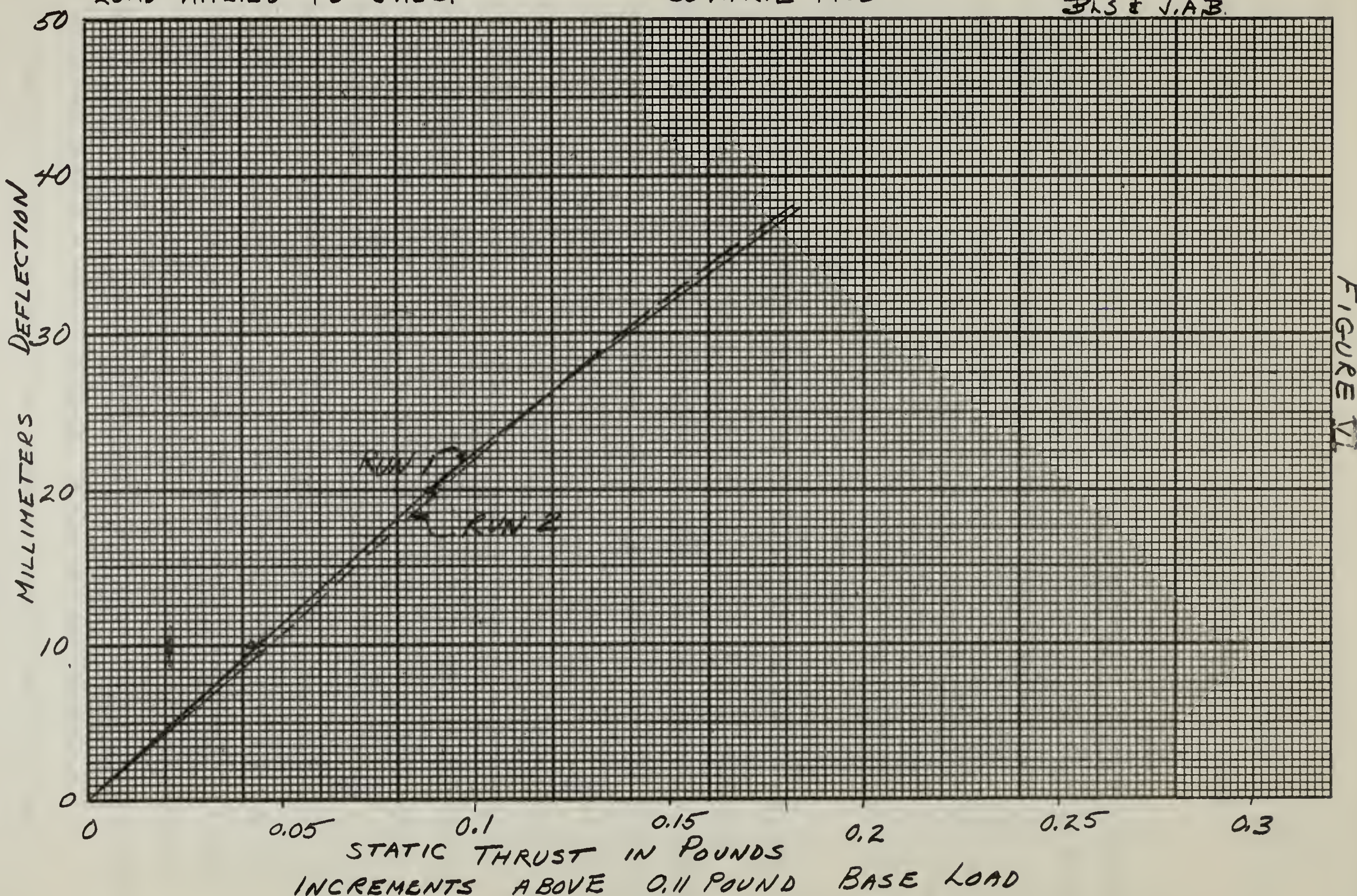


FIGURE V
STATIC CALIBRATION OF LVDT THRUST GAGE SHOWING UPPER & LOWER LIMITS OF READINGS (DASHED) AND CURVE THROUGH POINTS OF HIGHEST DENSITY (SOLID) DATA RECORDED BY MODEL 150 SANBORN DUAL CHANNEL RECORDER - X 20 ATTEN.

FIGURE VI

CALIBRATION OF LVDT THRUST GAGE FROM DATA RECORDED BY SANBORN MODEL 150 RECORDER "X20" ATTENUATION. RECORDER BALANCED WITH 1.1 POUND LOAD APPLIED TO GAGE. 30 APRIL 1956

BLS & J.A.B.



Primarily, it is felt, these effects are a direct product of the spring design and construction. With regard to linearity, it is known that the Schaevitz gage output versus core displacement is very linear in the range of motion involved in this design. Any nonlinearity is most probably a result of a nonlinearity in the spring constant.

The spring design assumes a linear spring constant. This assumption is in turn based upon certain construction requirements. The first of these requirements is that the springs must all be equally loaded at all times and must share equal increments of any load change. To attain this condition the springs must be of the same length, cross-section and must all have the same degree of fixity to the outer sleeve. Also entailed is the requirement that all springs have the same degree of fixity to the core shaft.

In the construction of the shaft and springs these conditions are not attained. The spring wire holes in the outer sleeve and core shaft are approximately 0.001 inch larger than the springs. This was done to facilitate insertion of the springs into the holes. When we consider that the Schaevitz gage produces full output for only 0.010 inch deflection then it is seen that large errors are likely to be introduced by even such a small tolerance. It is likely, therefore, that as the first load is applied only a portion of the springs are taking the load; hence the gage is operating at a relatively low spring constant. The high initial slope at the origin of the curve would substantiate this behavior. As the load is increased more springs are made to share the load, but not equally. The result is an increase in spring constant. Thus the central region of Figure V shows a decreased output for a given load increment. As the load is further increased all the springs are made to take part in the load resulting in a further increase in spring constant and hence reduced output. The region near 0.44 pounds load is seen to have less data spread than the central region. In the central region it is postulated that the loose springs have a random behavior which varies from run to run depending on the position of the spring relative to the sides of the spring holes. Some friction effects could occur until large deflections of the

core under higher loads force the springs to bear firmly against the hole walls. The fact that any given run falls on a fairly smooth curve differing from other runs is attributed to the fact that for any run the spring conditions are different (and depend on how the load was removed and also on how the load was applied).

In Figure VI are plotted the results of two runs made when the transducer was balanced with an initial load of 0.11 pounds. This was done to note the effect of load changes from a specified operating load point. The two curves are very close to one another and represent a difference of only 0.002 pounds for any given stylus deflection except near the higher end where the difference reached 0.005 pound. These curves also show the nonlinear nature of the gage.

Other possibilities for error arise from baseline drift and phase shift of the carrier frequency signal. It is felt that there errors are small and are not major factors. Baseline drift on the higher attenuation scales is hardly noticeable. Since no difficulty was experienced in balancing the gage, it is assumed that no large amount of phase shift was present in the system.

Figure VII shows the Lucite shaft torque gage and the thin ring thrust gage. The differential transformer thrust gage and the completed assembly of torque gage, thrust gage and drive shaft are illustrated in Figure VIII. Figure IX diagrams the proposed installation of the torque and thrust meter in the Series 60, Block 0.60 model.

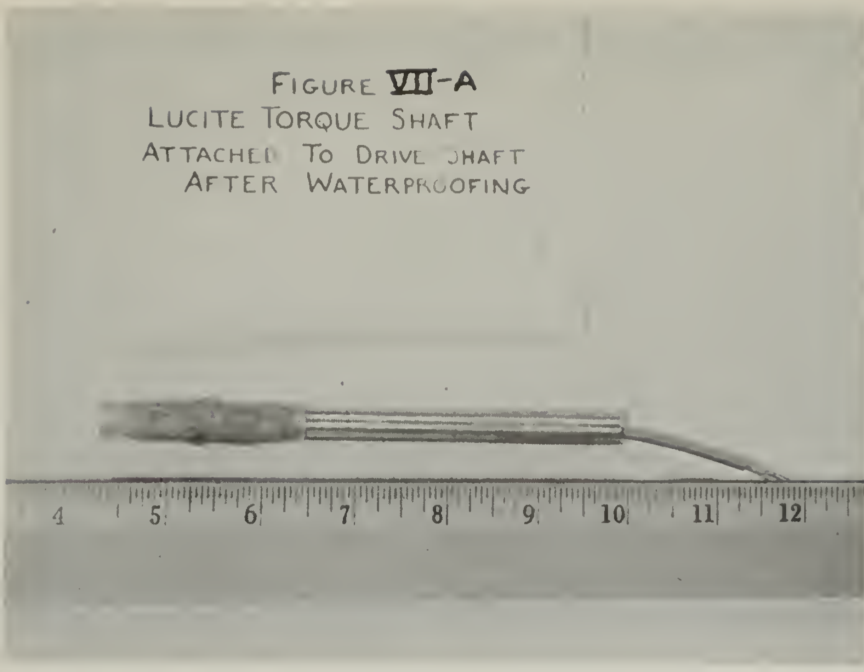


FIGURE VII-A
LUCITE TORQUE SHAFT
ATTACHED TO DRIVE SHAFT
AFTER WATERPROOFING

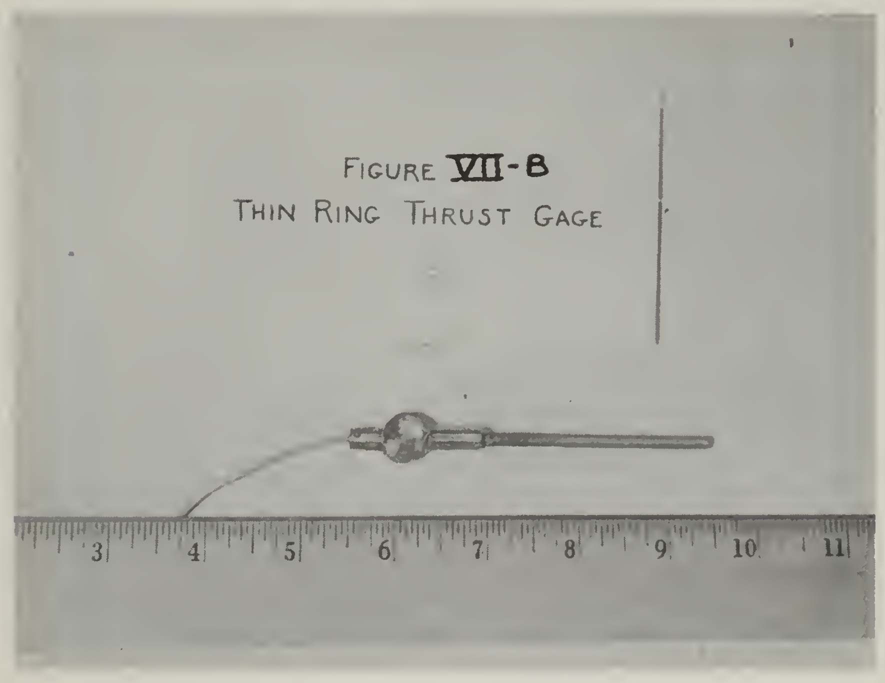


FIGURE VII-B
THIN RING THRUST GAGE

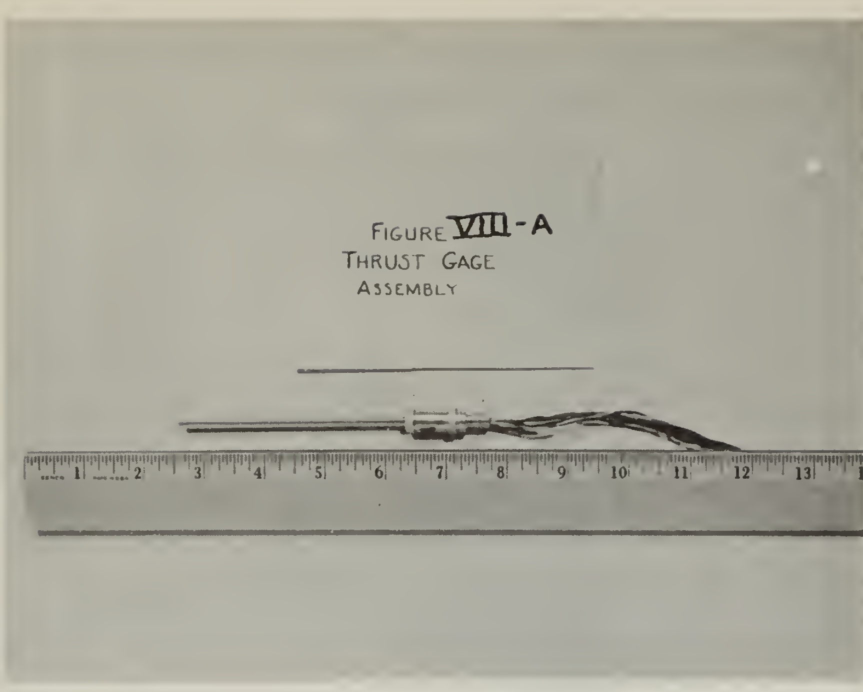


Figure VIII-A
Thrust Gage
Assembly

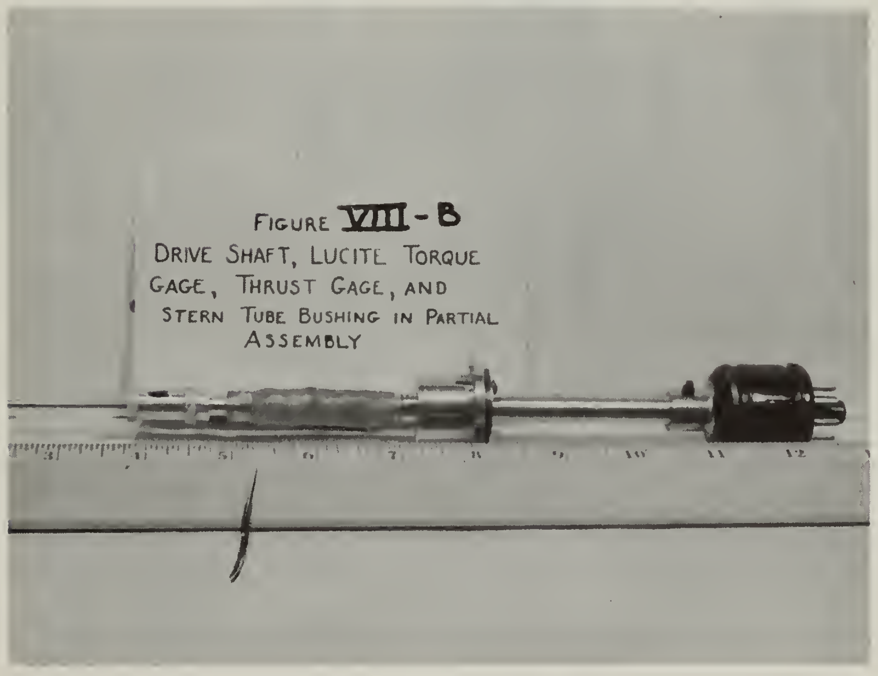


Figure VIII-B
Drive Shaft, Lucite Torque
Gage, Thrust Gage, and
Stern Tube Bushing in Partial
Assembly

FIGURE IX

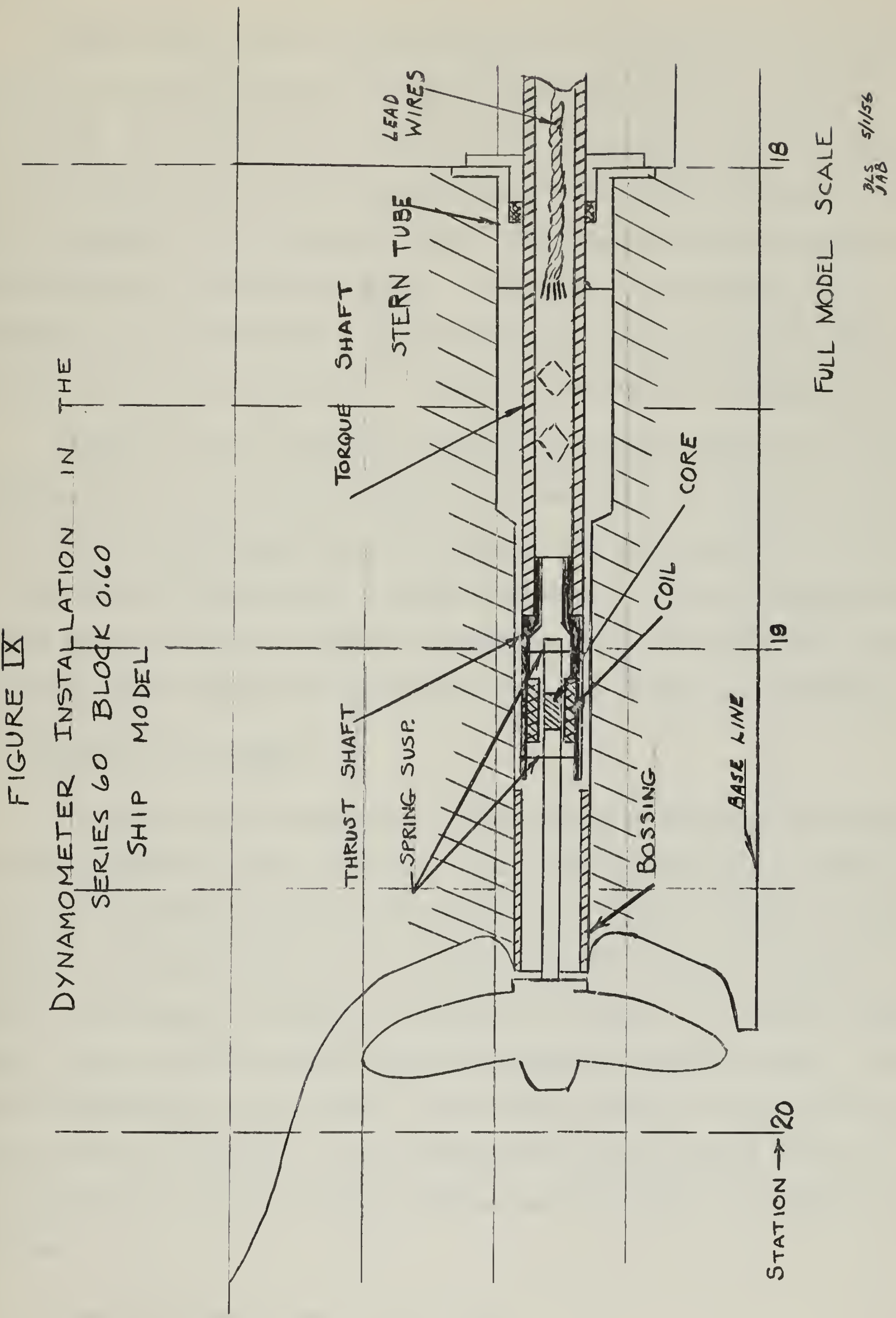


FIGURE IX
DYNAMOMETER INSTALLATION IN THE SERIES 60 BLOCK 0.60 SHIP MODEL

IV CONCLUSIONS

1. Torque Gage

On the basis of the results obtained we conclude that the Lucite shaft torque gage is found to be suitable for installation in a five foot ship model. Its small size, relatively high sensitivity, good linearity, and low bending response are considered to be excellent qualifications for its use in model self-propulsion tests. While the data spread found in the static tests was prohibitively large, it is felt that a calibration with a very small data spread would be obtained by immersing the gage in a large body of water at constant temperature, such as a towing tank, during calibration.

Although other torque measuring devices are available and might lend themselves to successful miniaturization, it is our opinion that the relative simplicity and ruggedness of the plastic and strain gage device developed are very desirable features. We conclude then, that if an acceptable signal to noise figure can be maintained by reducing slip-ring noise, the Lucite torque gage will best fit the application proposed in this investigation.

2. Thin Ring Thrust Gage

This gage has the advantages of excellent linearity in the range of interest, simplicity, and small size. The chief disadvantages of this gage are its low sensitivity and the lack of sufficient flexural rigidity. In this application whirling vibrations of large amplitude cannot be tolerated. In addition, high gage sensitivity is required to achieve an acceptable signal to noise ratio when the gage is used in conjunction with slip-rings. Because of the disadvantages noted above, and because the LDT thrust gage provides a much greater sensitivity, we conclude that the thin ring thrust gage is not satisfactory for the proposed application unless the ring design can be improved.

3. Differential Transformer Thrust Gage

This gage, as presently constructed, is unsatisfactory due to the large data spread obtained in successive runs. It is the opinion of the authors, as

stated in RESULTS AND DISCUSSION, that the primary source of data spread is an improper spring suspension. It is felt strongly that a great improvement in the design of the spring arrangement can be made with consequent large improvement in obtaining reproducible data. A discussion of such improvements is given in Chapter V, RECOMMENDATIONS FOR FUTURE RESEARCH.

We conclude that, because of the high sensitivity andsmall physical size, this type thrust gage is quite practical for use in ship model self-propulsion testing.

V. RECOMMENDATIONS FOR FURTHER RESEARCH

Based on our experiences in the design and construction of the various devices that have been discussed, and upon the results and conclusions presented, the following recommendations for future research are proposed:

1. The Lucite Torque Gage

(a) The present Lucite torque shaft should be calibrated while immersed in the towing tank or some other large body of water at constant temperature to determine if the large amount of base line drift persists under these conditions.

(b) In conjunction with (a) the present gage might be immersed in a container of water and the temperature varied to ascertain the unbalancing effects, if any, under such conditions.

(c) Investigations be made directed at determining the best manner of insuring accurate gage alignment on the shaft.

2. The Thin Ring Thrust Gage

(a) An investigation should be made to determine if any suitable materials with lower modulus than that of Brass, which was used in our gage, can be successfully employed; thereby realizing a greater output signal.

(b) Closely allied with (a) is the necessity of devising a more precise method of assembling a ring gage if the effects of dynamic unbalance are to be eliminated.

3. The Differential Transformer Thrust Gage

(a) A study should be made to determine a better method of spring suspension than the one used by the suthors. Several suggestions which we offer for what they may be worth are:

i. Filling the holes in the propeller shaft, around the springs, with some material such as solder or waterproof glue.

ii. Pre-load the present springs in some manner to insure that all springs are in intimate contact with the shaft before any thrust is applied to the shaft.

iii. Make a redesign of the assembly in which two thin diaphragms serve to support the shaft and act as springs. This would have the included advantage of excluding water from the shaft and allowing the buoyancy of the hollow shaft to partially counteract the effects of long shaft overhang.

(b) A point to be kept in mind when a redesign of the spring suspension is approached is that some means of construction should be found that would enable the forward set of springs to be examined after the gage is assembled. This is a defect in the present design that has been particularly annoying to the authors.

4. Slip-Ring Assembly

We recommend that for use in dynamic investigations a high quality slip-ring assembly be obtained. We feel that the items mentioned below would all be of importance in the fabrication of this assembly.

(a) The ring material should be silver.

(b) The brushes should be of the silver-carbon or gold-palladium type.

(c) Two more more brushes be employed on each slip-ring.

(d) The run-out in the rings should be as small as possible. A value of of 0.0002 inch maximum run-out was suggested to the authors by Mr. P.N. Bowditch of the M I.T. Instrumentation Laboratory.

(e) The bearing supports for the slip-ring assembly and the support for the stern-tube bearing of the model should be mounted on a common bed-plate. This would allow drilling the mounting holes for all bearings with a single machine set-up. It is felt that this would afford the best possible alignment of all elements of the drive shaft which are located inboard in the model.

5. Miscellaneous Recommendations

(a) Although the eight-pin cable connector is an ideal shaft coupling in principle, because it serves to make both electrical and mechanical coupling, it is not very satisfactory in use. The primary reason is that the metal housing fits onto the plastic base by spring action. The housing is free, therefore, to shift position and such a shift causes misalignment of the shaft. Another housing which could be attached to the base by some positive mechanical means such as a set screw would be superior. Perhaps some other type of multiple conductor cable connector might be available that would be satisfactory,although the authors did not succeed in locating one.

(b) In order to minimize difficulties arising from run-out in the completed assembly of drive shaft, torque shaft, and thrust gage, it is recommended that some method for insuring that these three components are in perfect alignment be adopted. One possibility which the authors would suggest is that these sections be threaded on a lathe before strain gages or the differential transformer are mounted. In this way the sections could be joined by screwing one into the other. If necessary, a final truing cut could be made onthe assembly.

(c) Looking into the future, and anticipating the construction of a wholly satisfactory torque and thrust dynamometer such as we have investigated here, the authors suggest that some thought be given to the choice of a drive motor and the design of a feed-back control system for maintaining the propeller speed of the model constant during self-propulsion testing.

APPENDIX

APPENDIX A. DETAILED PROCEDURE

1. General

In the introduction several devices were listed that theoretically may be used to develop the torque and thrust sensing devices desired. An investigation into the details of each device led to the selection of electrical resistance strain gages. Resistance strain gages are relatively inexpensive, can be obtained in sizes as small as 1/16 inch (grid dimension), have a high degree of precision, and are relatively adaptable to most applications with regard to arrangement. In addition, the literature cites numerous cases where these gages have been successfully utilized in the measurment of torque and thrust on high speed rotating shafts. The theory applicable to these gages is well established and may be found in great detail in reference (3). On the basis of these considerations it was concluded that resistance type strain gages were a good starting point for the investigation.

2. Preliminary Investigation of the Use of Electrical Resistance Strain Gages and Suitable Shaft Material

a. Having decided on a suitable type of gage the problem of finding a suitable shaft material had to be resolved. The primary effect of the material lies in its modulus of elasticity. This can be readily seen by examination of Hooke's law:

$$s = Ee \tag{1}$$

where s is the stress in pounds per square inch in the outer fibers of the material to which the gage is applied, E is the modulus of elasticity of the material in pounds per square inch, and e is the strain in inches per inch generated in the outer fiber by the applied stress. Since the gages are sensitive to strain it follows that for a given stress the strain-gage output will be inversely proportional to E.

The stress, in turn, is generated by a thrust load which, in this design, is limited to approximately 0.25 pounds. The stress is determined, in the case of thrust on a shaft, by the relationship

$$s = P/A \tag{2}$$

where P is the load in pounds and A is the cross-sectional area, in square inches, of the shaft in a plane perpendicular to the applied thrust. Combining equations (1) and (2) we arrive at the relationship

$$E = P/Ae \tag{3}$$

P is fixed as stated above. In addition, A is limited by the space available in the model due to the fineness of the stern section (Figure I). A high value of strain is desired. Signals generated by strains of the order of 1000 micro-inches per inch are usually desired in full scale applications. With such signals noise inherent in the electrical system is relatively insignificant.

A further limitation arises from the fact that, even with very small gages, the necessary arrangement of the gages to measure thrust or torque limits the shaft diameter to a minimum of approximately one-quarter inch for thrust, and three-eighths of an inch for torque. The reasons for this limitation will be discussed in greater detail in section 3. If we consider the above limitations and assume that $P = 0.25$ pounds, $A = \frac{\pi D^2}{4}$ where $D = 0.25$ inches, substitution into Equation (3) yields

$$E = 4 \times 0.25/0.25^2 \pi e = 5.08/e \tag{4}$$

or

$$e = \frac{5.08}{E}$$

Examination of Equation (4) shows that in order to produce appreciable strain, E must be as low as possible. For example if steel were used ($E = 3 \times 10^7$ pounds/in^2) the calculated strain would be $e = 5.08/3 \times 10^7 = 1.69 \times 10^{-7}$ in/in. Such a small strain can be detected by the gage but would be completely lost in noise.

It is concluded that, for the thrust shaft, a material of low modulus of elasticity must be used. In fact, if the acceptable signal level is set at 100 to 200 micro-inches per inch, Equation (4) shows that E must lie between 2.54×10^4 and 5.08×10^4 pounds per square inch.

An alternative to using a material with such low E would be to reduce the section area, A, to a minimum by the use of a hollow shaft. For example, consider the use of an aluminum shaft ($E = 10^7$).

$$e = P/AE = 0.25/A \times 10^7$$

$$\text{where} \qquad A = \frac{\pi}{4}(D_o^2 - D_i^2)$$

D_o = outside diameter of shaft and
D_i = inside diameter

If we let $D_o = 0.25$ inch and assume a desired strain of 200 micro-inches per inch, we can solve for the required thickness of the shaft wall, t . This value is found to be 0.00015 inch . This is of such small magnitude as to preclude practical construction, necessary strength, and ruggedness. It should be noted that even if the acceptable strain were reduced to 20 micro-inches per inch, the thickness, t, required would be 0.0015 inch which is also considered impractical.

b. A similar set of calculations may be made for the torque shaft. The relationships involved are slightly different. The maximum stress developed in a circular shaft under pure torsion is given by

$$s_{max} = M_t r/I_p \tag{5}$$

where M_t is the applied torque, r is the radius of the shaft to the outer fiber, and I_p is the polar moment of inertia of the shaft. For a hollow shaft Equation (5) may be written as

$$s_{max} = \frac{16\ M_t}{\pi d^3 \left[1 - \frac{d_i^4}{d^4}\right]} \tag{6}$$

where d_i is the inside diameter of shaft, d the outside diameter of shaft. This stress occurs along an axis at 45 degrees to the shaft axis. To calculate the strain developed by this stress revert to equation (1). Substitution of Equation (6) into Equation (1) yields

$$e_{max} = \frac{16\ M_t}{\pi E d^3\ \left(1 - \frac{d_i^4}{d^4}\right)} \tag{7}$$

Calculations in Appendix (C) for Series 60 Block 0.60 indicate a maximum model torque of approximately one inch-ounce. Using this value of torque, aluminum for shaft material, and an outside diameter of three-eighths of an inch, the inside diameter may be expressed as a function of the maximum strain, e maximum

$$d_i = 0.375 \sqrt[4]{1 - \frac{1.887}{\pi e_{max}}} \tag{8}$$

where e_{max} has the units of micro-inches per inch.

If the acceptable strain is set at 200 micro-inches per inch, equation (8) gives an inside diameter of 0.3741 inch . Thus, the wall thickness is 0.001 inch. Further computation will show that the use of a practical wall thickness would reduce the strain to an undetectable value. The conclusion reached at this point is that the use of electrical resistance strain gages requires a shaft material whose modulus of elasticity is significantly lower than that of aluminum.

c. On the basis of the conclusion stated in 2b , attention was directed to the selection of a shaft material with a low modulus of elasticity. In this connection it was felt that plastic materials might be suitable. Rubberlike, low rigidity materials were automatically excluded because of the low flexural strength, their high creep and their associated high hysteresis characteristics. A material with high flexural strength is particularly

[illegible]

$$[illegible]$$

[illegible]

$$[illegible]$$

[illegible]

[illegible]

[illegible]

desirable to minimize the propeller deflections resulting from the rather large overhang anticipated in the design. As mentioned in the introduction, it is desired to have no bearings on the propeller shaft aft of the dynamometer. Mock gage arrangements on a dummy shaft showed that the expected overhang would be approximately 4 inches. While this length is twice that needed for the gage installations, it was necessitated by space limitations in the stern section of the model.

Certain general requirements were set up with regard to the properties desired in a plastic material to be used in this application. These are:

a) Low Modulus
b) Moderate to high flexural strength
c) Low creep characteristic
d) Good machinability
e) Uniform properties
f) High resistance to water, grease, and solvents
g) Ability to be bonded to paper gages
h) Little change of properties with aging

On the basis of tabulated characteristics Vinylidene Chloride (Saran) was chosen. This plastic satisfied most of the above requirements. Saran has an extremely high water resistance, very good machinability, moderate flexural strength, and changes very little with age. The stated modulus of elasticity varied widely from source to source and lies in the range of 0.3×10^5 to 1.7×10^5 pounds per square inch and the properties are uniform. The chief disadvantages of using Saran are, first, like most low modulus materials, it has a high creep rate compared to metallic materials, and, second, it is difficult to bond to any other material. These disadvantages were outweighed by the low modulus of elasticity and the other desirable properties mentioned previously.

Calculation of the strain which would be developed in a solid 0.25 inch diameter rod of Saran by a 0.25 pound axial load reveals that we can expect

[illegible]

[illegible]

[illegible]

[illegible]

to obtain a strain of 102 micro-inches per inch using an assumed modulus of 0.5×10^5 pounds per square inch. As will be shown later, a suitable thrust measuring strain gage arrangement would double this output thus giving an equivalent output of approximately 204 micro-inches.

Similarly a calcuation for strains due to pure torsion as outlined in section 2 b indicated that we could expect a maximum strain of 100 micro-inches per inch. This torque calcuation is based on a solid 0.4 inch diameter Saran shaft loaded with one inch-ounce of torque.

The conclusion was reached that a Saran torque and thrust shaft dynamometer would be constructed and tested. The tests would be directed towards checking the signal outputs versus load, investigation of creep, and determination of bending effects. Investigations would also be conducted to determine a suitable bonding cement for bonding the resistance strain gages to the Saran shaft.

3. Gage Installation and Construction of Saran Dynamometer Shaft

a. Brief Electrical Strain Gage Theory

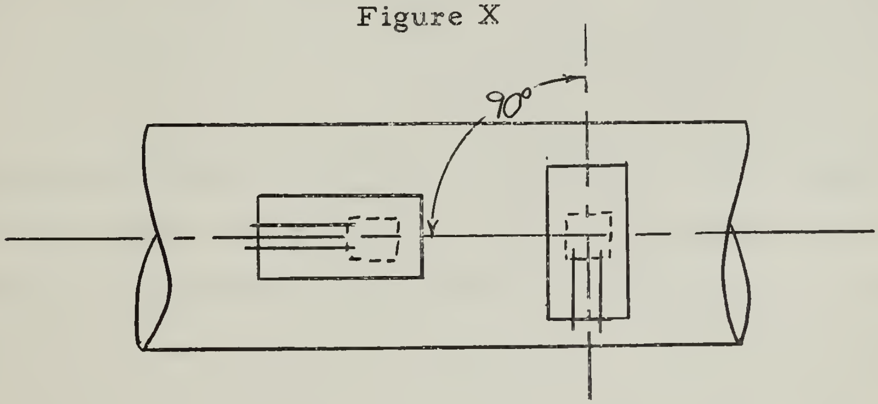
A shaft loaded with pure axial thrust develops a strain in the shaft material along the longitudinal axis. Due to the Poisson effect, there is associated with this longitudinal strain a smaller induced transverse strain. A resistance grid properly bonded with its axis parallel to the shaft axis will be strained with the shaft. The strain in the wire causes its resistance to change proportionately. For a given input voltage to the grid, the change in resistance causes a current change inthe wires of the grid. Measuring this current change with the aid of amplification provides a means of measuring changes in strain. Since the strain is proportional to the load which causes it we have then a measure of the applied load.

If the load is composed of bending as well as thrust, the strain developed is a combination of the two effects. In order to isolate the bending component, a second gage is placed diametrically opposite the first gage. Thus

the strain due to bending is equal in both gages but of opposite sign. Addition of the two signals then automatically cancels the bending component. The thrust components, having the same sign, will add and thereby double the signal output.

Normally two other gages are added to the thrust arrangement to form a four arm bridge. The latter gages are arranged with their grids transverse to the shaft axis. By doing this these gages are made insensitive to thrust and bending and are, for this reason, called dummy gages. The primary function of the dummy gages is to accomplish automatic temperature compensation. This is necessary since temperature changes in the shaft would cause strains to be generated as a result of expansion and contraction of the shaft material. The dummy gages, when mounted on the same material and in the same temperature environment will be strained equally. By placing the four gages in a bridge arrangement, as shown in Figure X the temperature effects can be made to cancel. Principal stresses caused by pure torsion consist of a tensile stress at 45 degrees to the shaft axis and a compressive stress oriented at 90 degrees to the tensile stress. Thus, there is no axial component of stress in the shaft. For this reason there is no net transverse stress. Hence, the thrust arrangement shown in Figure X is insensitive to torsion.

Figure X



Thrust Gage Arrangement on Circular Shaft

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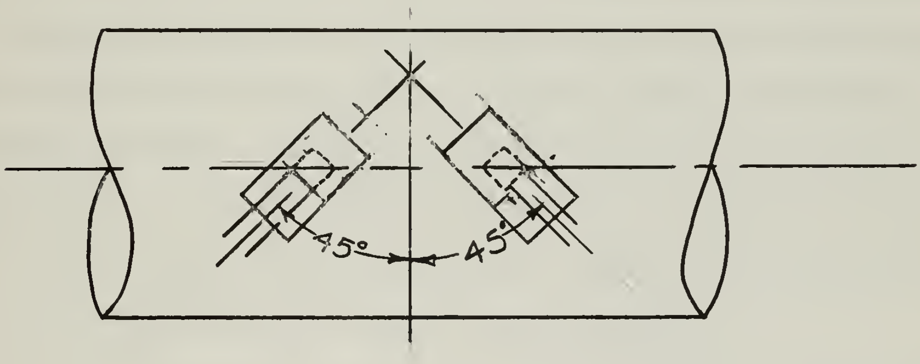
The same principles permit a gage arrangement for the measurement of torque independently of thrust and bending. Consider four gages mounted as shown in Figure XI. Two gages are mounted on one side of the shaft and oriented with the grid axes at an angle of 45 degrees with respect to the shaft axis and at an angle of 90 degrees with respect to each other. Diametrically opposite to these two gages, two other gages are mounted in a similar arrangement. Any torsion that is applied to the shaft causes maximum strain along the gage grid axes. Since, in a four arm bridge, changes in opposite arms add, while changes in adjacent arms subtract, it can be seen that the output of the bridge can be made equal to four times the output of a single gage. This is possible because the strain generated in two of the gages have the same sign while the gages oriented at 90 degrees to the first two have strains of equal magnitude but of opposite sign to that of the first two. If the gages of same sign are placed opposite to one another in the bridge arrangement their signals will add while the subtraction of the adjacent arms amounts to addition due to the sign difference. A detailed discussion of this bridge arrangement is contained in Reference (3).

Since thrust will generate equal strains (of the same sign) in all arms, no net signal is produced. Likewise bending strains will be cancelled. In principle, the arrangement shown in Figure XI is highly satisfactory. It must be remembered that the above theory depends on several important factors. The first is that the gages must have exactly the same response. The gage factors and the resistance of the gages must be single valued among the four gages comprising the bridge. In practice this is very nearly met. Gages of 120 ohms are guaranteed to be within plus or minus 0.3 ohm of the specified value in any given gage pack. Gage factors are guaranteed to be within plus or minus two percent of the prescribed value. Variations in the gages, therefore, would yield very minor effects.

The second factor of importance is the gage alignment. Theoretical predictions are based on exact angles and diametral positioning of the gages. Deviations from exactness have probably the largest effect in causing deviation from the theory and hence result in having some response to bending, temperature, torque (in the case of the thrust gage) and thrust(in the case of the torque gage).

Other effects may arise as a result of unequal bonding, differences in cement layers under the gages, local stiffening due to cement on low modulus materials, and lead wire capacitance.

Figure XI



Torque Gage Arrangement on Uniform Circular Shaft

For a detailed mathematical development of bridge theory, see Reference (3).

b. Selection of Electrical Resistance Strain Gages

The SR-4 type strain gages (for details see reference 10) have a wide variety of application. They can be installed on surfaces of various shapes, have linear response, and very small mass. These gages are available in more than 100 varieties thus insuring suitability for a specific

application. The principal variables involved in gage construction are:

1. filament wire material
2. base carrier material
3. bonding cement
4. filament construction
5. lead wire connections

Combinations of the construction variables will produce the numerous SR-4 gage varieties. These gages vary with regard to overall and grid dimensions, grid patterns, resistance, and gage factor. Filament material is considered to be the most important variable. It determines the gage factor or the change of resistance of the filament (grid) for a unit change in strain. It also determines the temperature sensitivity of the gage. SR-4 gages employ two materials, one is a copper nickel alloy and the other is an isoelastic material. The gage factors for these materials are 2 and 3.5 respectively. These values should be taken as approximate only. Actual values vary from gage to gage and are determined by the manufacturer. Values are furnished with the gages when purchased.

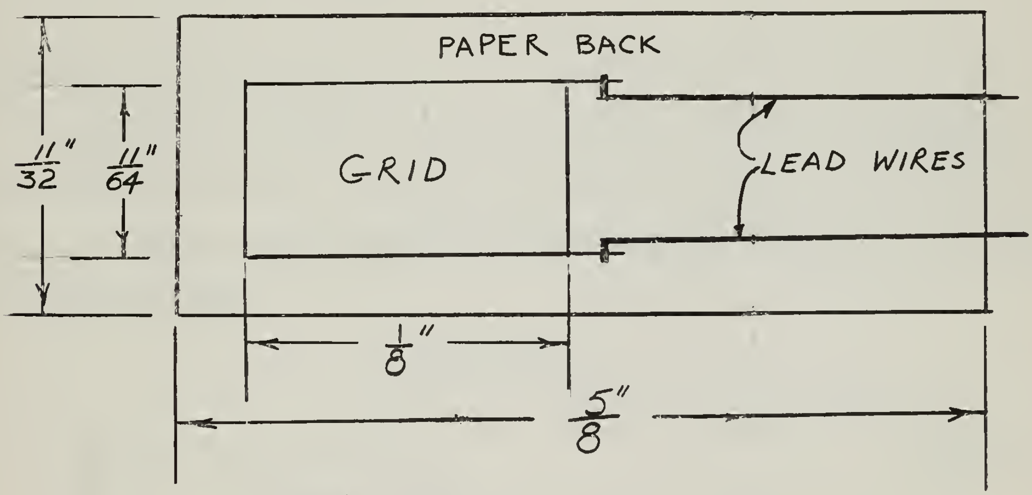
In most cases where strains are to be compared under varying loads a stable reference is required. Iso-elastic gages, while having high gage factors, are extremely temperature sensitive and usually cannot be temperature compensated easily. The copper-nickel alloy, on the other hand, has a lower sensitivity to temperature and is easily compensated particularly for small temperature deviations. The disadvantage of the copper-nickel lies chiefly in its lower gage factor when compared with iso-elastic materials.

On the basis of these facts, the gage selected for this design was a copper-nickel or A type. Within the A category are many sizes. The size range varies from 1/16 inch to several inches for the grid dimension. Associated with the size of the gage is the resistance of the gage. Larger gages have

more or longer loops in the grid and hence higher resistance. Generally speaking, a high resistance value is desirable for two reasons. One reason is that recorders used to record strain indications, such as the Sanborn Model 150 Dual Channel, will operate at reduced sensitivity if the transducer impedance is less than 100 ohms. The second reason is that the output signal of a higher resistance transducer is affected less by undesirable external effects such as slip ring noise. This is an important consideration for cases where signals generated by transducers on rotating shafts must be removed by a slip ring arrangement.

Considering all the factors mentioned, the authors selected an SR-4 strain gage, type A-8. This gage has a catalogue resistance value of 120 ohms and a gage factor of 1.72. Gages as purchased were rated at these values by the manufacturer*. Resistance value is guaranteed by the manufacturer to be within 0.3 ohm of the specified value. Gage factors are given a tolerance of plus or minus 2 percent. The A-8 gage is shown in Figure XII.

Figure XII



SR-4, A-8
Electrical Resistance Strain Gage

* Baldwin-Lima-Hamilton Corporatinn

c. Bonding of Gages

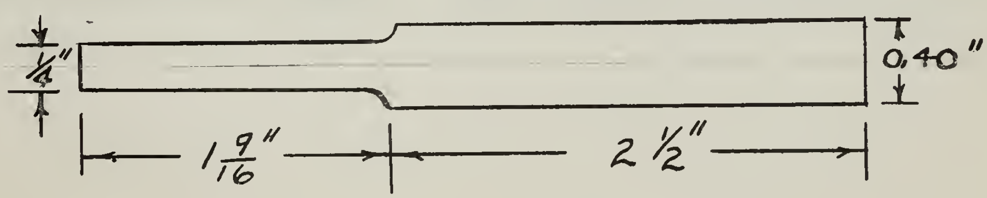
The bonding of paper backed gages to metal surfaces is described in detail in Reference (11). When bonding to metal a nitrocellulose cement such as Duco is normally used. Specimens of Duco were placed on a Saran rod and baked at 125 degrees Fahrenheit for several hours. The bonding was found to be ineffective inasmuch as the cement was easily peeled off after 24 hours of drying. As a result of this simple test, the authors consulted with the Baldwin-Lima-Hamilton Corporation branch at Cambridge, Mass. This firm recommended the use of an epoxy resin which is normally used on etched foil gages. Cementing is achieved by mixing 8 to 10 parts by weight of Ciba Company hardener number 951 to 100 parts by weight of Ciba company's 502 Araldite resin. Instructions specify that where a chemical balance is not available the mixture can be approximated by using an eye dropper and adding 7 drops of hardener to the 3 gram kit bottle of 502 resin. The pot life of the cement after mixing is approximately thirty minutes.

Test specimens were made by adding the cement mixture to a Saran rod followed by several hours of baking and 24 hours of drying. The cement proved to form a successful bond.

d. Construction of Saran Shaft

In preparation for gage installation a Saran shaft was machined to the dimensions illustrated in Figure XIII. The dimensions were chosen on the basis of a dummy model made by mounting paper cutouts, representing gages, on a wood rod.

Figure XIII



"Saran Dynamometer Shaft Dimensions"

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[illegible]

[illegible]

[illegible]

[illegible]

4. Gage Installation Procedure

a. Gage Trimming

In order to form a suitable arrangement of the gages on the shaft it was found that some of the excess paper backing on the gage had to be trimmed off. Extreme care had to be exerted in the trimming process in order to prevent damage to the delicate grid and lead wires. In addition, judicious trimming involved leaving sufficient paper particularly near the lead wires to support the leads during handling and installation. The connection between the lead wires and the grid is very sensitive and easily ruptured. Approximately 1/16 inch was trimmed around the periphery of the gages.

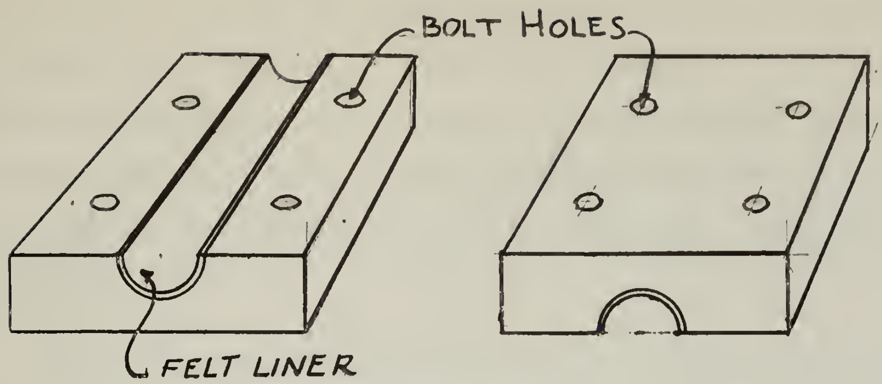
b. Forming Gages

To facilitate installation on the shaft the trimmed gages were preformed by pressing them lightly against the Saran shaft until they had acquired some curvature. Here, in particular, great care was required since bending of the lead wires might rupture the grid connection. After forming, the gages were checked for electrical continuity and resistance values.

c. Application Jigs

It is necessary that the gage be kept under pressure while the cement is drying. For this purpose two special jigs were fabricated. These consisted of wooden blocks whose faces contained grooves drilled out to the outside diameter of the shaft. The grooves were lined with felt padding to distribute pressure when the blocks were clamped around the shaft. The two halves were provided with bolt holes so that tightening up on the bolts would apply pressure to the gages. A typical jig is illustrated in Figure XIV.

Figure XIV



Gage Application Jig

d. Cleansing

The Saran shaft was prepared by lightly rubbing the surface with 00 emery paper followed by cleansing with a swab soaked in carbon tetrachloride to remove all soil. The gages, held by the lead wires, were swirled in carbon tetrachloride and permitted to dry.

e. Marking Shaft for Gage Location

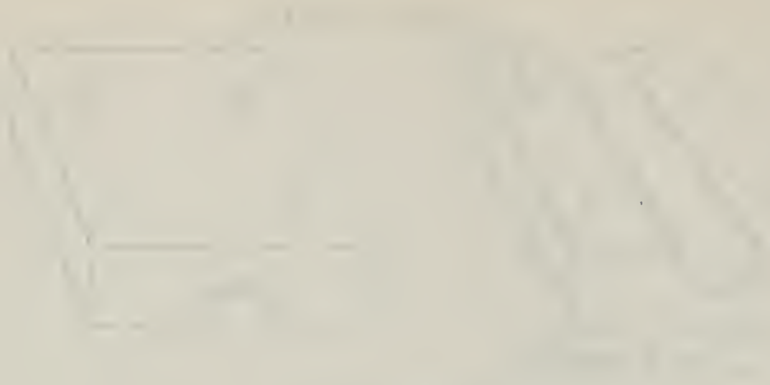
Positioning of the gages was provided for by carefully marking points on the shaft with a light scribe. Longitudinal and diametral positions were determined on a bench lathe,

f. Cementing Procedure

Epoxy cement mixture was applied to the gage backing and to the Saran shaft. The gage was pressed lightly into place by hand while a covering of wax paper was wrapped around the shaft and gage to prevent bonding of the cement and felt pressure padding. The jig was then applied and tightened to apply pressure to the gage during the drying period. The assembly was then baked at 125 degrees Fahrenheit for two to three hours followed by air drying for twenty-four hours. The jig was then removed and the gages were permitted to dry for another twenty-four hours before testing.

g. Comments on First Installation

In the first installation the above technique was followed but the four thrust gages were applied simultaneously in an effort to reduce installation



time. After drying, the installation was found to be out of line. Two of the thrust gages had slipped around the shaft during the application process. The misalignment amounted to a shift of the gages of 1/16 of an inch from being diametrically opposite. The two misaligned gages were stripped off the shaft and two more were applied in their place. The results of the second alignment were better than the first but there was still visible misalignment.

The primary difficulty involved in the alignment lay in the fact that it was difficult to hold the gage in position while wrapping the wax paper around it. During this stage the cement is still very fluid and any slight disturbance moves the gage from its desired position. In addition, the gages cannot be maintained in position while fastening the jig on the shaft.

Despite the above difficulties, the torque mounting was made following a similar procedure with the exception that the gage bridge was preformed on wax paper. The gages were held to the wax paper by melted wax. This assembly was then wrapped around the torque shaft to which cement had been applied. This device greatly facilitated positioning of the gages when compared to the primitive method used for the thrust assembly. The wrapping around, however, proved to be inadequate and presented a serious problem because of its bulk and rigidity. The arrangement resulting from this device was fairly good but the unaided eye could easily find angular as well as lateral and peripheral misalignment.

It was decided to test the two installations before any further revisions were made, in order to determine two things which were of interest at the time. The first bit of information desired was the effect that misalignment of the gages would have on the bending response of the torque and thrust shafts. The second item of interest was to see if the magnitude of the output signals agreed with the values previously calculated for assumed loads.

The four gages on each shaft were connected in a bridge arrangement. The bridges were formed on the shaft, rather than bringing the leads

out and making external connections, for two reasons. The first reason was that the number of slip rings required was cut in half by this arrangement. The second reason was that any unbalancing effects due to the slip rings would be minimized in this way.

Connection wires were soldered to the four corners of the bridges, led through small holes drilled in the shaft and out through the hollow center of the shaft to the driving end.

5. Static Tests of the First Dynamometer

To implement testing of the thrust section of the dynamometer the shaft was mounted in a wooden block drilled out to fit snugly around the torque section of the shaft. The torque gages were padded to prevent accidental damage. The wooden block was firmly attached to a larger wooden base. A terminal strip was mounted on this base and the lead wires protruding from the hollow shaft were attached to this strip. The excitation and signal leads were carried from here to a Sanborn, Model 140, single channel amplifier and recorder by shielded cables. A load platform was placed on the upper end of the thrust section to support precision weights.

With the platform in position and with no load applied the bridge was balanced. Weights were then added to the platform and recorded on the recorder tape. After completing several tests in thrust, the dynamometer was mounted in the torque testing device illustrated in Figure XV. One end of the shaft was held fixed by a set screw. The other end was cemented to a ball bearing supported live center to which a torque lever arm was attached. By hanging precision weights on the hook at the end of the arm, known torques were applied to the shaft. To minimize friction effects the bearings were cleaned thoroughly and lubricated and the loads were dithered. Strain signals were recorded on tape with the single channel Sanborn recorder mentioned previously.

The results of these first tests pointed out several important facts. In the thrust test it was determined by calculation that for 0.25 pound

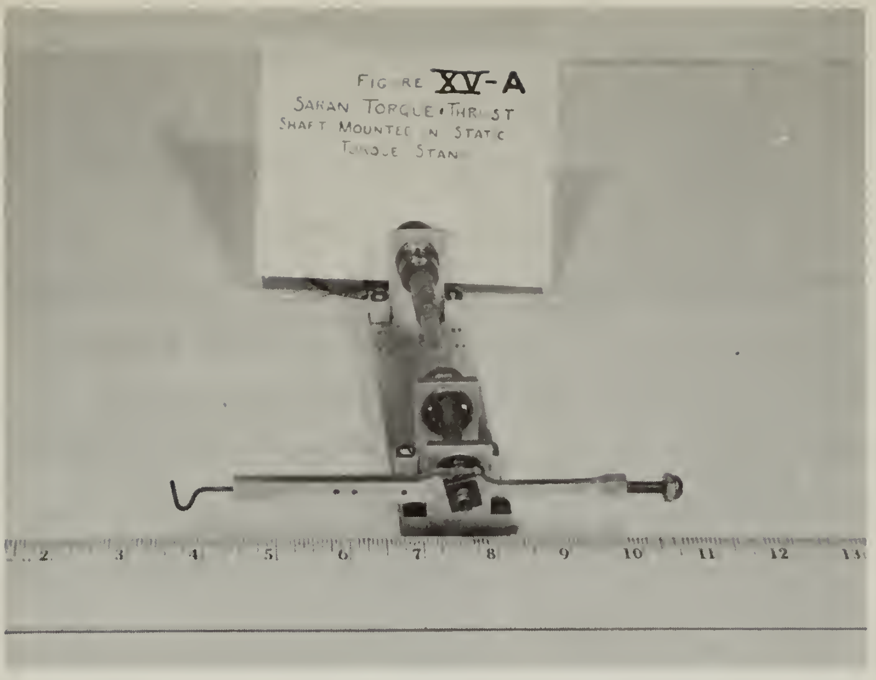


Figure XV-A
Saran Torque & Thrust
Shaft Mounted in Static
Torque Stand

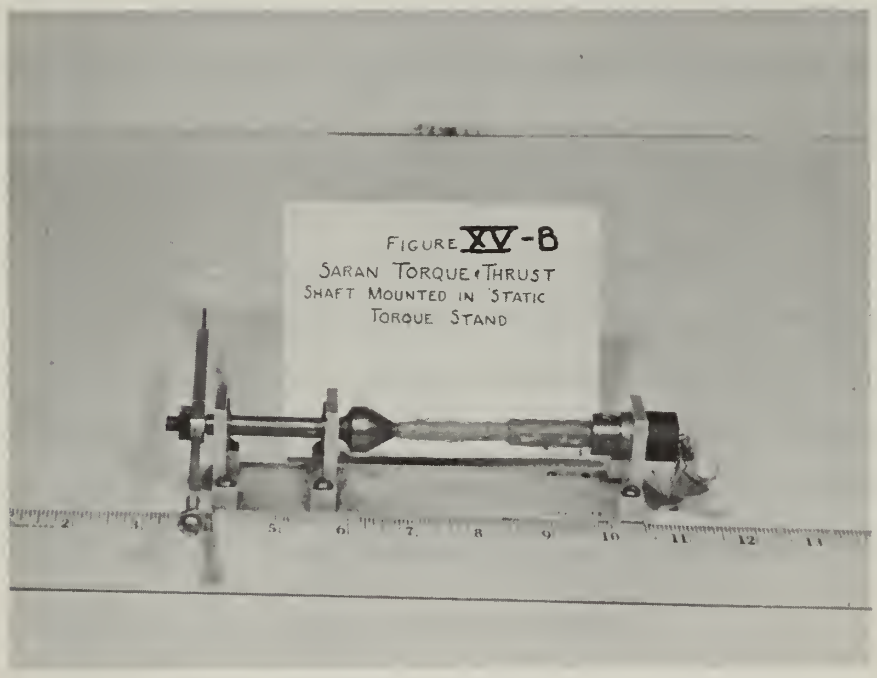


Figure XV-B
Saran Torque & Thrust
Shaft Mounted in Static
Torque Stand

axial load the strain in the shaft was 36 microinches per inch as compared to 100 microinches per inch predicted. Details of the calcuation are given in Appendix C. Another observation made was that application of the load was followed by creep which lasted approximately 20 seconds. When a load of 0.25 pounds was applied to the shaft, for example, the initial stylus deflection (attenuator set at X1) was 15 1/2 millimeters. After 20 seconds the deflection had reached a nearly steady value of 20 millimeters. Thus it is seen that the creep has a large magnitude and relatively long duration. The creep phenomenon observed is held to be unsatisfactory since it would preclude observation of thrust variations at blade frequency. Also, the creep duration would mean that a model with this dynamometer installed would have progressed over half the length of the tank before the creep would be essentially completed.

On the basis of creep alone it was decided to abandon this first dynamometer model. Before abandonment, however, crude tests were made to determine the bending response of the meter by application of light transverse loads to the end of the shaft. The load used was that of a 2-inch model propeller weighing approximately one ounce. Large responses were noted as a result of this test. This confirmed our belief that misalignment was an extremely important factor in the design and that suitable steps had to be taken to provide very accurate positioning of the gages.

The torque section was tested in the test stand and it was found that with one inch-ounce of torque applied a deflection of 15 millimeters of the stylus was obtained when the recorder amplifier was set at X1 attenuation and maximum sensitivity. These results also indicated a marked decrease in expected output. As in the case of the thrust shaft large bending response was noted and the creep phenomenon was again experienced.

The results obtained clearly indicated that:

a) Saran was an unsuitable material because of its high creep.

b) A more suitable method must be devised for mounting gages.

c) The output of the gages was less than anticipated.

The probable cause of result (c) is the stiffening effect of the gages and cement on the Saran shaft. Reference (6) discusses the stiffening effect of gages mounted on materials of low modulus. The gages and materials discussed are different from those utilized in the dynamometer but the conclusion that stiffening effects of two or three times may be experienced is equally applicable here. In order to alleviate the large creep and stiffening effects it was decided to experiment with a plastic of higher modulus of elasticity. Although smaller strains would be expected due to the increased modulus, it was our hope that we could use another cement which would have a much smaller stiffening effect. In this way we might yet realize a greater output signal than that obtained with the Saran shaft. In addition, it was felt that the creep phenomenon would be considerably reduced in magnitude.

6. Examination of Lucite Shaft Material

a. General

Lucite has a modulus of approximately 5×10^5 pounds per square inch or about ten times the value for Saran. Its machinability is fair compared to Saran and its resistance to water and other solvents is not as good as Saran. Duco cement contains a solvent which dissolves lucite. This latter feature was used to advantage in positioning the gages on the shaft.

b. Construction

Two hollow Lucite tubes were obtained in the 1/4 inch and 3/8 inch size. The 1/4 inch tube (1/8 inch inside diameter) was used for the thrust mounting. It was contemplated that when a successful model of each measuring device was obtained they would be joined by inserting the 1/4 inch tube into the end of the 3/8 inch tube. Lead wires would be run through the center of the tubes.

The thrust gage positions were accurately scribed using a lathe. For the torque arrangement a square and diagonals were drawn on paper. The sides of the square were made equal to the circumference of the torque

shaft; thus, the diagonals would form opposite 45 degree helixes on the shaft. The paper was wrapped around the shaft with two sides parallel to the shaft axis which was scribed on the shaft. Holding the wrap-around with scotch tape a sharp pointed scribe was used to mark points through the paper on the shaft. In this manner an accurate 45 degree helix was impressed on the shaft, Diametral positinns were then marked to locate the gage centers.

c. Gage Mounting

The same procedures described in section 4 were used, with a few exceptions. The first exception was the use of Duco cement to mount the gages. The second exception was that the gages were mounted one at a time, clamped, and permitted to set for about two hours after which another gage was mounted and the assembly was reclamped in the pressure block. This was done until four gages were mounted for each configuration. The procedure described is unorthodox but was resorted to in an effort to obtain some results in a reasonable time.

Duco cement softens Lucite and to some extent this is undesirable. This property was advantageous, however, inasmuch as it permitted one to apply a gage with hand pressure. After several minutes the cement and softened Lucite had set sufficiently to hold the gage in position. The amount of set was sufficient so that minor disturbances did not shift the position of the gage appreciably. This feature permitted the successful application of wax paper and the wood jig without causing excessive gage shift. An added advantage of this process over the epoxy resin process is that the Duco cement requires no mixing or special precautions and is easily and cheaply procured.

It was noted that it was more difficult to machine the Lucite than Saran and that caution had to be observed to prevent cracking of the Lucite shaft. Lucite is best machined at high speed using small cuts and a small cutting surface.

d. Testing

Computations similar to those in section 2 of this Appendix indicated that the strain would be theoretically reduced by a factor of ten from the value computed for Saran. In order to obtain appreciable signal output with a 0.25 pound thrust load the cross section of the thrust gage had to be materially reduced. Attempts at reducing the cross section to a wall thickness of ten thousandths of an inch were made; but, in each case, the Lucite cracked in the process. As a consequence the section could not be reduced sufficiently to produce appreciable output of the thrust gage.

The torque mounting proved to be satisfactory. The gage was mounted in the torque testing stand and connected to a strain gage recorder; calibration runs were made by the addition of weights to the torque arm. In this test several determinations of the transducer gage factor were also made. Tests were directed at determining the gage sensitivity to bending. This was accomplished by the addition of a 1.1 pound weight to the torque shaft at a point two and three quarter inches from the point of shaft support thus generating a bending moment of 3.025 inch-pounds. It should be pointed out that this load is larger than any anticipated load on the shaft by at least a factor of ten. Even with this large load the recorder stylus deflected only 12 millimeters with X1 attenuation, full sensitivity, gage factor set at 2 and the sero suppression out. Such a deflection is very small when compared with the deflections obtained in response to torques in the operating range.

The bending moment test was conducted after the thrust and torque shafts had been assembled and inserted into the dynamic test stand. Figure XX shows the complete test setup with recording instrument and dynamometer as wired for testing.

7. Slip Ring Assembly

A slip ring assembly of some sort was a necessity since the electrical signals had to be removed from the rotating shaft for connection to the measuring instruments. For best results, a commercially designed and

constructed assembly would have been desirable. Unfortunately, lack of time and money precluded having a commercial assembly built. It was felt, however, that since this thesis was in the nature of a feasibility study, a "home-built" set of slip rings could be made which would enable us to determine whether the basic torque and thrust gage sections would do the job they were designed to do.

For ease of construction we decided to employ an insulating material for the slip ring shaft if possible. A piece of 1/2 inch outside diameter bakelite tubing was chosen for the shaft. This tubing had an inside diameter of 1/4 inch which was adequate for insertion of metal end-sections to serve as bearing journals. The 1/4 inch inside diameter also gave a reasonable amount of space for feeding the lead wires in from the rings. After some deliberation on the method of attaching the lead wires to the rings, it was decided to make the connection by shrink fitting the rings over the lead wires. A slight groove was turned in the bakelite at the point which was to be the mid-point of each ring. The lead wire was then inserted through a hole into the hollow center of the shaft and several turns of wire were wound in the groove and soft soldered together to form a band. The slip rings were then heated until they had expanded enough so that they could be slipped down the shaft, over the wire-solder band. When the rings cooled they were shrunk down tightly upon the wire thus making both mechanical and electrical contact. Since the rings were quite hot when placed on the shaft there was some local charring of the surface of the bakelite shaft. This was minimized by slipping the rings into position quickly and by quenching the assembly in water as soon as each ring was put into its proper position.

Silver tubing would have been preferred for ring material, but at the time the proper size could not be readily obtained; so expediency dictated the use of copper which we had at hand. The copper rings were made 1/4 inch wide and 1/16 inch thick. The thickness was made this great in order to insure that there would be ample material for taking several cuts on the rings in a lathe for the prupose of truing the assembly.

The selection of brushes for the assembly was again a matter of expediency. A number of phosphor bronze relay contact arms with silver contact points were available. These were mounted on a strip of 1/4 inch Plexiglass plastic and mounted to bear firmly, but lightly, on the rings.

The bedplate and two bearing supports of the assembly were fabricated from pieces of 1/4 inch Plexiglass. This material was chosen chiefly for the purpose of keeping the weight of the assembly as low as practicable.

As mentioned previously, there was some surface charring of the bakelite shaft when the rings were installed. The presence of this char was deemed to be undesirable and was removed by taking a light cut on the shaft, between rings, on a lathe. Since this cut left the surface of the bakelite without the protective glaze which it had previously, it was felt that something should be done to prevent this unprotected surface from absorbing moisture and accumulating dirt. A layer of ordinary red sealing wax was melted onto the shaft covering the spaces between rings and over the rings themselves. This layer was then turned down to the level of the rings and final truing cuts taken on both rings and wax. This procedure gave a flush surface consisting of alternate copper rings and wax-filled spaces. Final smoothing of the rings was done by turning the slip ring assembly at approximately 2000 r.p.m. and pressing a strip of 6/0

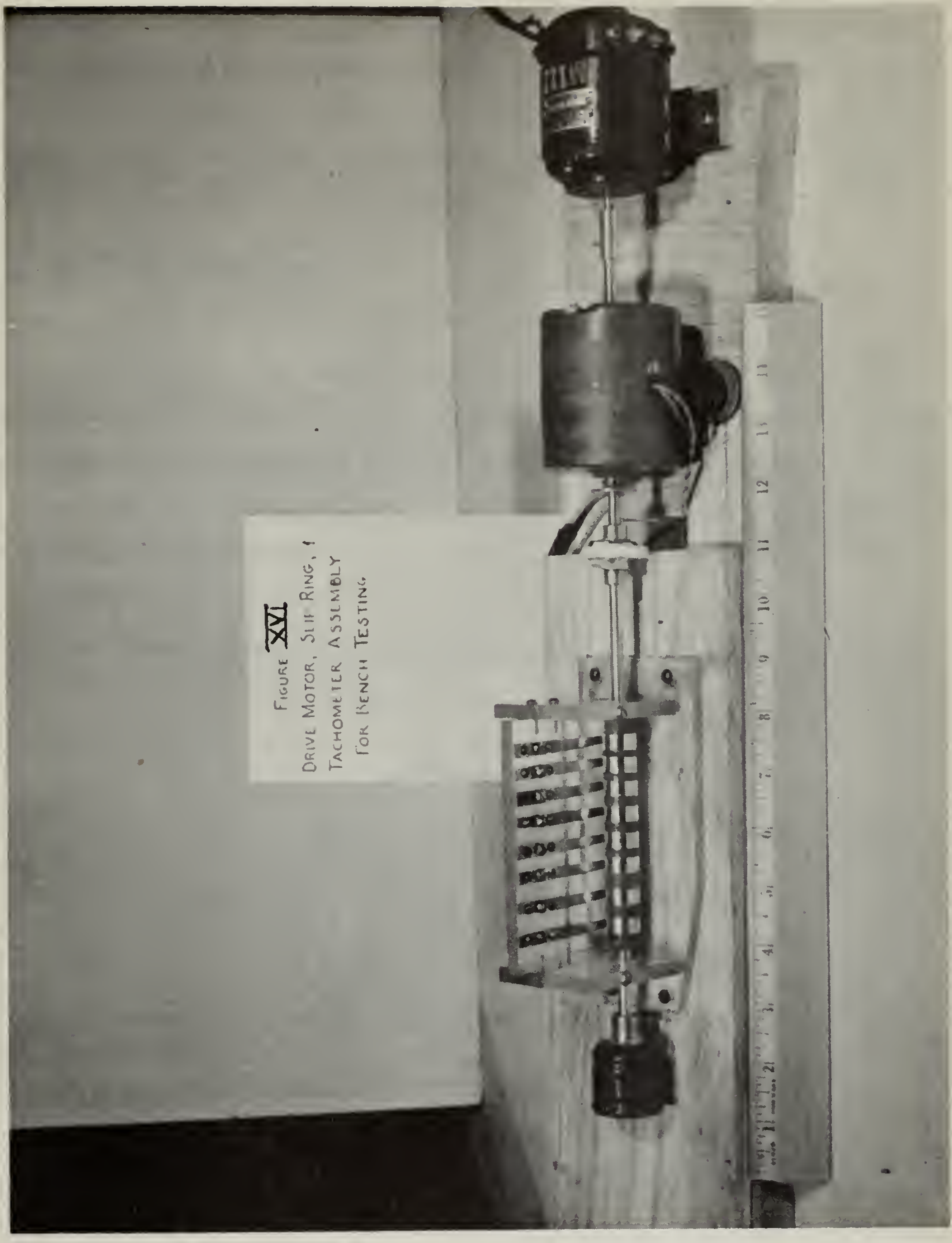


FIGURE XVI
DRIVE MOTOR, SLIP RING, &
TACHOMETER ASSEMBLY
FOR BENCH TESTING

sandpaper against the rings with a wooden block which had been drilled out to fit the contour of the rings.

The completed assembly is shown in Figure XVI mounted with the drive motor and tachometer used as a bench test stand. The lead wires were brought out through a hole inthe metal shaft to an eight pin cable connector which served as both an electrical and mechanical coupling. The completed assembly had an insulation resistance of 10 megohms or more between any pair of rings.

8. Thin Ring Thrust Gage

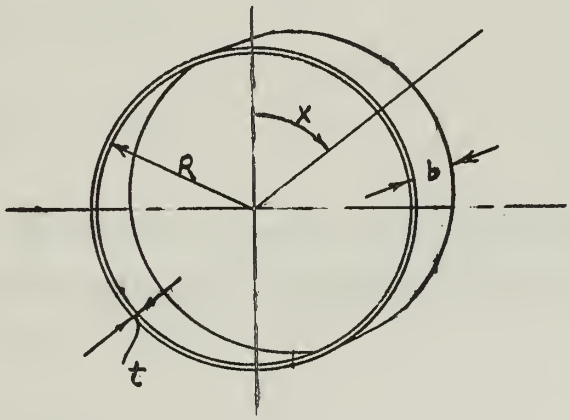
a. Description

A new attempt was made to construct a suitable thrust gage. The device described in this section is essentially a 'proving ring'. By mounting four resistance strain gages on a thin circular ring it is possible to obtain a gage which will indicate thrust. The gages are mounted at quadrature to the thrust or shaft axis.

b. Theory

The following formulae aretaken from Reference (12)

Figure XVII



The bending moment M at any point in the thin ring due to an axial load W is given by,

$$M = WR(0.3183 - 1/2\ Z) \tag{13}$$

where $Z = \sin X$.

From equation (13) the maximum positive moment and maximum negative moment are given by

$$M\ (pos) = 0.3183WR \quad \text{at } X = 0 \tag{14}$$

$$M\ (neg) = -0.1817WR \quad \text{at } X = 90 \text{ degrees} \tag{15}$$

The tensile load, T, at any fiber is

$$T = -1/2\ WZ \tag{16}$$

When X is 90 degrees, $T = -W\ 1/2$. Stresses due to tension and bending set up in the ring at $X = 90$ degrees are given by

$$s_t = T/A \tag{17}$$

where A is the cross section area of the ring; and

$$s_b = My/I \tag{18}$$

where y is one half the ring thickness and I is the moment of inertia of the ring cross sectional area about its neutral axis.

From equations (17), and (18), and Hooke's law

$$e_t = s_t/E = T/AE \tag{19}$$

$$e_b = s_b/E = My/IE \tag{20}$$

The strains may be written in terms of the ring parameters by substituting into Equations (19) (20), Equations (15) and (16); using t/2 for y, bt for A and $\frac{bt^3}{12}$ for I. Then

$$e_t = -\frac{-W}{2\ bt\ E} \tag{21}$$

$$e_b = -\frac{0.1817\ W\ R\ t\ 12}{2\ bt^3 \quad E} = -\frac{1.090\ \ W\ R}{bt^2\ \ E} \tag{22}$$

Reasonable ring parameters for this design are:

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

b = 0.25 inch

R = 0.1875 inch

t = 0.010 inch

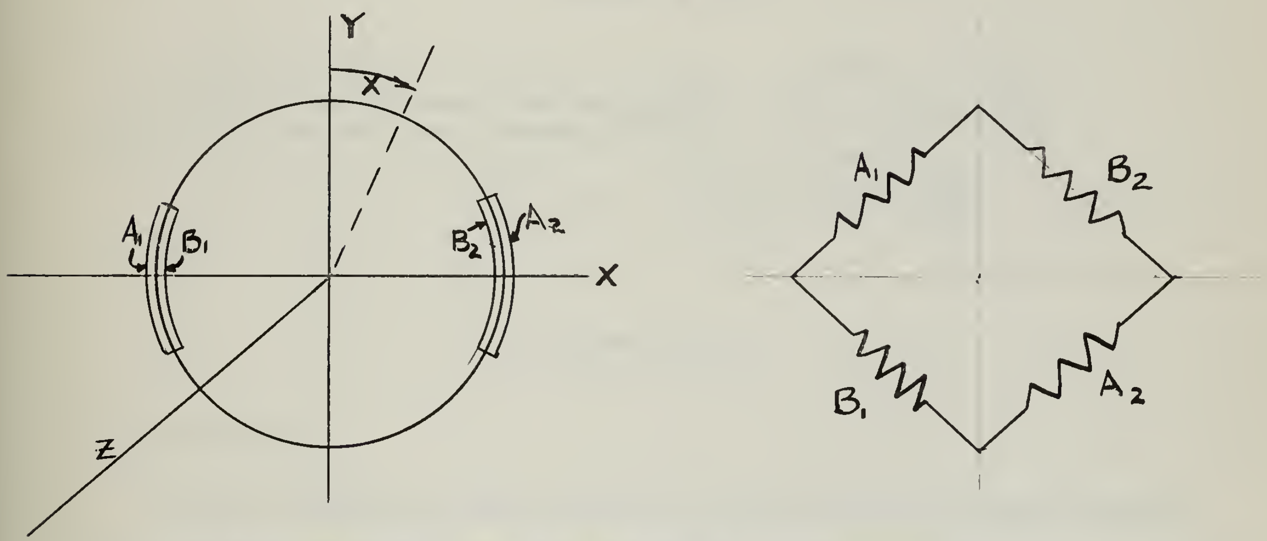
E = 14×10^6 (for rolled brass) pounds/in^2

W = 0.25 pounds

Substituting of these values into Equations (21) and (22) we obtain values of e_t and e_b equal to 3.57 and 145 micro inches per inch respectively. Thus, it is seen that the strain due to tension or compression is small compared to that due to bending.

By a suitable bridge arrangement as shown in Figure XVIII this computed output may be quadrupled.

Figure XVIII



Thin Ring Thrust Gage and Bridge Arrangement

If the ring is subjected to bending in the yz plane the strains in the direction of the grid axes will be small since the grid axes are oriented at 90 degrees to the principal stress axes. Bending in the xy plane will generate principal stresses in the direction of the grid axes; however, since the algebraic sign of the strain generated in A_1 is opposite to that generated in A_2, the addition of the two effects a cancellation. The same applies to strains generated in B_1 and B_2. Theoretically, then, the thin

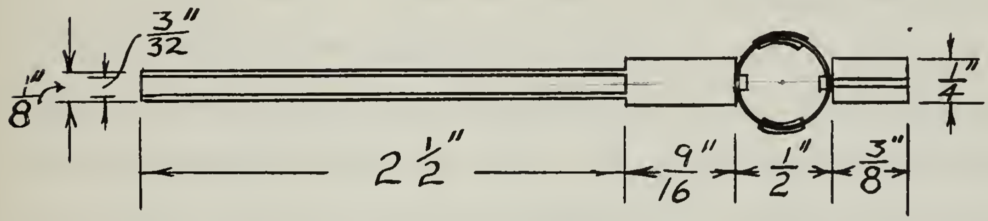
ring should be insensitive to bending in any plane.

c. Fabrication

A ring thrust gage was fabricated and is illustrated in Figures VII-B and XIX. The ring and shaft are made of rolled brass. The shaft pieces are hollowed to reduce weight and also to provide a path for the bridge lead wires. Four SR-4, A-8 strain gages were installed with the aid of specially constructed jigs. Duco cement was used for bonding.

The shaft pieces were silver soldered to the ring. The ring was drilled to receive the shaft piece ends which were reduced in diameter to attain a reasonable approximation to point contact.

Figure XIX



Thin Ring Gage

d. Calibration

A static calibration was run on the ring gage by placing precision weights on the end of the long shaft piece. For this calibration a Sanborn Dual Channel Model 140 recorder and amplifier was used. The gage was also spun at approximately 2000 rpm to determine what whirling effects were present. It was found that a large amplitude of deflection resulted. This effect is the primary drawback of the thin ring gage. The cause of the whirling effect can be traced to the fact that the thin ring does not have sufficient strength in flexure. In order to approximate point contact between the shafts and the ring one must accept low flexural strength.

Because of this large disadvantage to the ring gage no formal bending tests were made. Rough tests were performed by applying light loads to the

[illegible]

[illegible]

end of the long shaft piece while the short end was held fixed. The gage was found to have some bending response but this response was not of sufficient magnitude to impair the thrust indications desired.

9. Linear Differential Transformer Thrust Gage

a. Description and elementary theory

As a result of the failure to obtain satisfactory results with strain gages for thrust measurement, it was decided to try a linear differential transformer as a transducer for this application.

The linear differential transformer (LDT) consists of three coils equally spaced on a cylindrical coil form. A magnetic core in the form of a rod is positioned axially inside the coil and provides a path for magnetic flux linking the coils. When the center coil is energized with alternating current, voltages are induced in the two end coils. These two coils are connected series opposing so that the net output is the difference of the voltages induced in these two coils. When the core is positioned exactly in the center of the assembly this difference voltage will be zero.

When the core is displaced from the center position, the voltage induced in the coil towards which the core is moved increases. The voltage induced in the other coil decreases. The result is a net differential voltage output from the transformer. By proper design of the coil assembly, this voltage can be made to vary linearly with displacement of the core. If the core is displaced in the opposite direction from the central position a similar differential voltage output is obtained with a 180° difference in phase.

The advantages of the LDT for this application are its small size and weight, the negligible actuating force required, and the relatively large voltage outputs that can be realized for small displacements. For more detailed information on these devices the reader is referred to References (21), (22), (23) and (24).

b. Description of thrust gage mounting

Since the space available for the thrust gage was quite limited, a Schaevitz Engineering, series "M", type 010M, gage was chosen. This gage has external dimensions of 5/16 inch outside diameter by 7/16 inch long. The bore is 1/8 inch and the magnetic core is 0.108 inch in diameter and 0.210 inch in length. The core is threaded on both ends with a 1-72 female thread for attachment to the actuating device.

Because the action of the gage is dependent upon axial motion of the core, a method of suspension had to be found which would allow the propeller shaft to move axially in response to thrust. One method by which this might be achieved would be the insertion of a flexible coupling in the shaft between the drive and the propeller shaft. The coil would be fixed to the hull and would not require slip rings for removal of output signals. This proposal was rejected for two reasons:

a. A long overhang of shafting of the diameter of the magnetic core (0.108 inch) would have been necessary, with the likelihood of extremely severe whirling of the propeller.

b. If there was any end play in the bearings of the drive motor, it would be read as a thrust.

Although the use of slip rings to remove the signals from the LDT was necessitated, another mode of suspension was decided upon which would shorten the overhanging length of small diameter shaft and eliminate false indications of thrust due to end play in the drive shaft assembly. The proposed arrangement is shown in Figure IX.

The LDT coil was inserted in an aluminum tube which was machined to an outside diameter of 3/8 inch to conform to the outside diameter of the torque shaft. The propeller shaft (with the magnetic core inserted therein) is suspended by two sets of four cantilever springs at right angles to one another. One set of springs supports the propeller shaft at each end of the coil. These springs serve the dual purpose of permitting axial movement of the

propeller shaft while transmitting the torque from the small diameter propeller shaft to the 3/8 inch outer tubing. It can be seen that with this arrangement any displacement of the shaft due to end play will be accompanied by an equal displacement of the coil and hence there will be no output signal developed due to such a displacement.

c. Design of the spring suspension.

Since a longitudinal displacement of 0.010 inch produces full output from the transformer, we considered the springs as simple cantilevers to determine the size necessary to limit the deflection to this amount with the anticipated thrust of 0.25 pound applied.

Consider a single cantilever which we will assume to be fabricated from steel piano wire. The maximum deflection is given by the relation

$$y = \frac{P\,l^3}{3\,EI} \tag{23}$$

where P = transverse load applied at free end, pounds;

l = length of cantilever, inches;

E = Modulus of elasticity, pounds per sq. inch;

I = Moment of inertia of spring cross section.

For a spring fabricated from circular wire, the moment of inertia is given by

$$I = \frac{\pi D^4}{64} \tag{24}$$

where D is the wire diameter in inches.

Substituting Equation (24) into Equation (23) and solving for the necessary wire diameter one obtains

$$D = 1.615 \left(\frac{P\,l^3}{yE}\right) \tag{25}$$

If we now substitute into this result the maximum deflection desired (0.010 inch), the modulus for steel (30×10^6), the load absorbed by one spring (1/8 x 1/4 pound) and the length of the cantilever (1/2 x 5/16 inch)

[illegible]

the allowable diameter of the spring wire is found to be 0.00721 inch. In order to allow for some overload, the diameter of the wire actually used was made 0.010 inch.

These calculations ignore the effects of spring stiffening due to the transmission of torque and the fact that the springs are not truly cantilevers but are fixed to some extent where they are held by the propeller shaft. These effects and the effect of propeller shaft weight will be compensated for by actual calibration of the system in use.

d. Fabrication of Differential Transformer Thrust Gage

Several attempts at fabrication of the thrust gage whose design has been described were made by the authors. These attempts all failed due to the fact that a high degree of precision is called for in lining up the holes for the springs in the inner and outer shafts. Through the interest of Professor J.F. Reintjes, the aid of the M.I.T. Servomechanisms Laboratory was enlisted and a model of the thrust gage was assembled. This is the gage shown in Figure VIII A.

e. Testing the LDT Thrust Gage

Static tests were made on the LDT thrust gage by mounting it in a vertical position with the propeller shaft portion pointing upward. A small, light, block of wood was drilled out to fit snugly over the end of the propeller shaft. This piece of wood served as a scale platform upon which weights were placed to simulate the thrust which would be developed by a propeller in service. The differential transformer was connected to a Sanborn Model 150-1100 carrier preamplifier. This preamplifier, together with its associated recorder constitutes all of the necessary instrumentation.

Dynamic testing of the LDT thrust gage was to be carried out following assembly of the drive shaft, torque shaft and LDT thrust gage into one unit. This complete unit is shown in Figure VIII B. Due to a lack of time

this dynamic testing was not carried out. The assembled shaft unit was mounted in the test stand, however, and the torque meter was connected and balanced with the motor stationary and no load applied to the shaft. When the drive motor was energized the slip ring noise that was generated was sufficient to completely swamp the recorder, even when the attenuator was advanced to the X200 setting. In addition to this undesirable effect, it was discovered that there was considerable run-out in the assembled shaft and no further tests were made due to likelihood of damaging some part of the assembly by continuing to turn at high speeds while so much dynamic unbalance existed. The apparatus as assembled for testing is shown in Figure XX. The schematic wiring diagram for the test arrangement is given in Figure XXI.

FIGURE XX
Dynamic Test Stand Assembly
Connected to Sanborn Model
150 Dual Channel Recorder

FIGURE XXI

SCHEMATIC OF DYNAMIC TORQUE AND THRUST APPARATUS

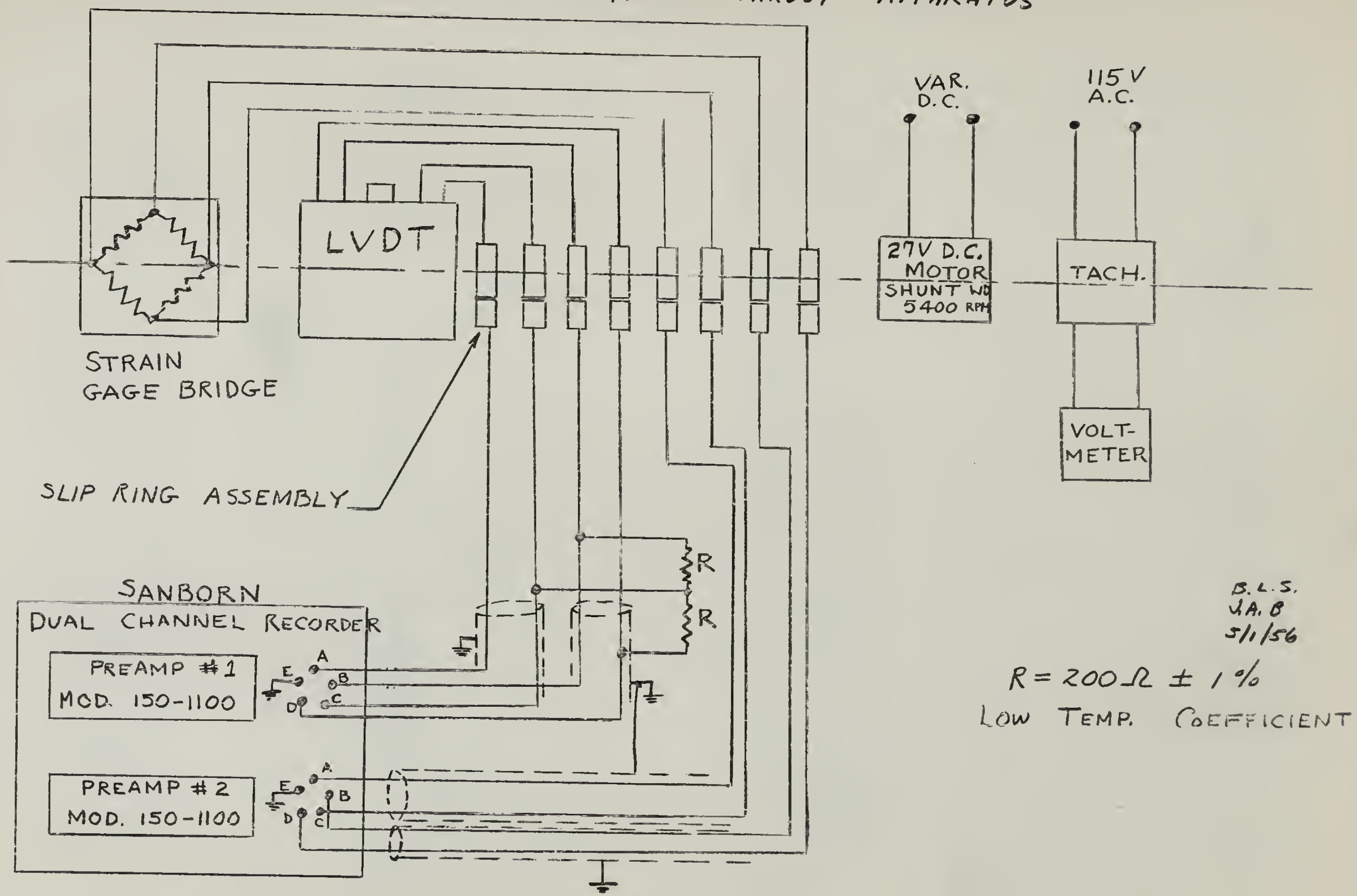
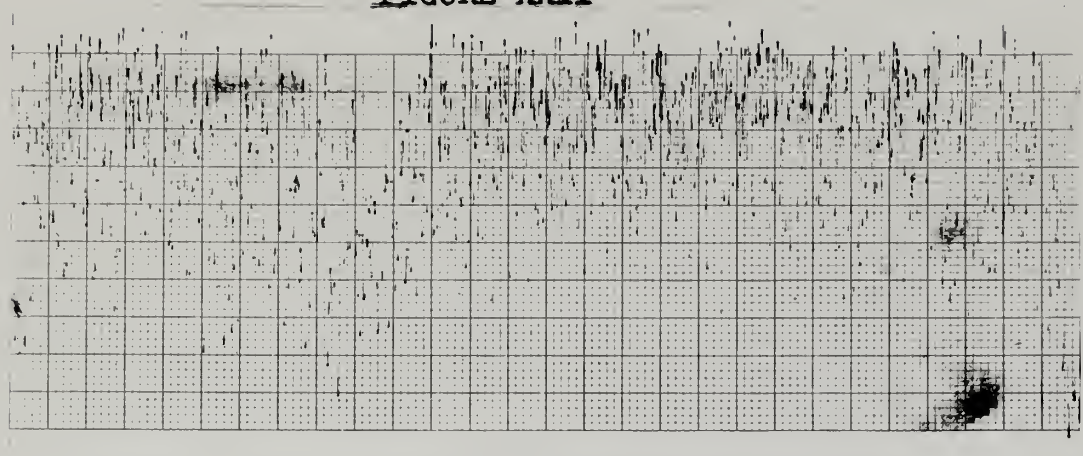
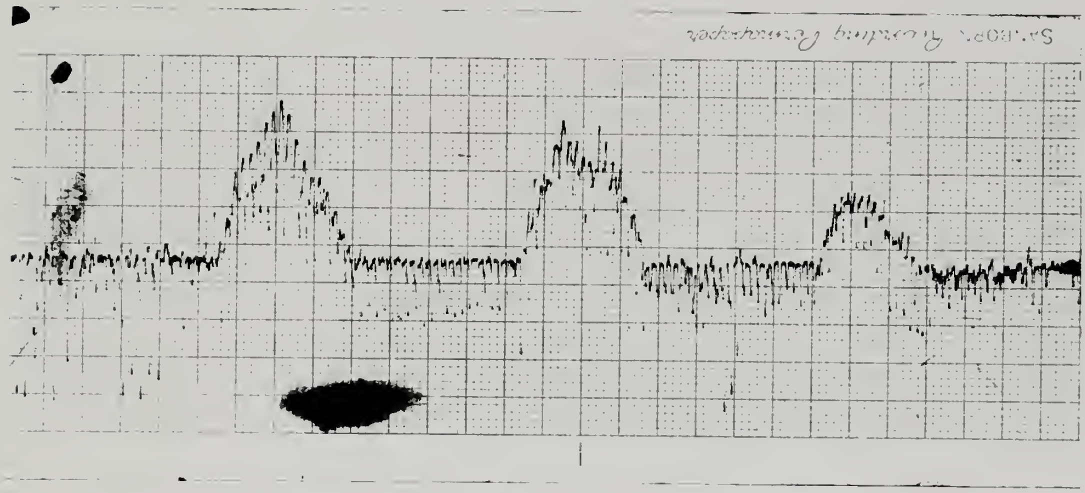


FIGURE XXI

FIGURE XXII



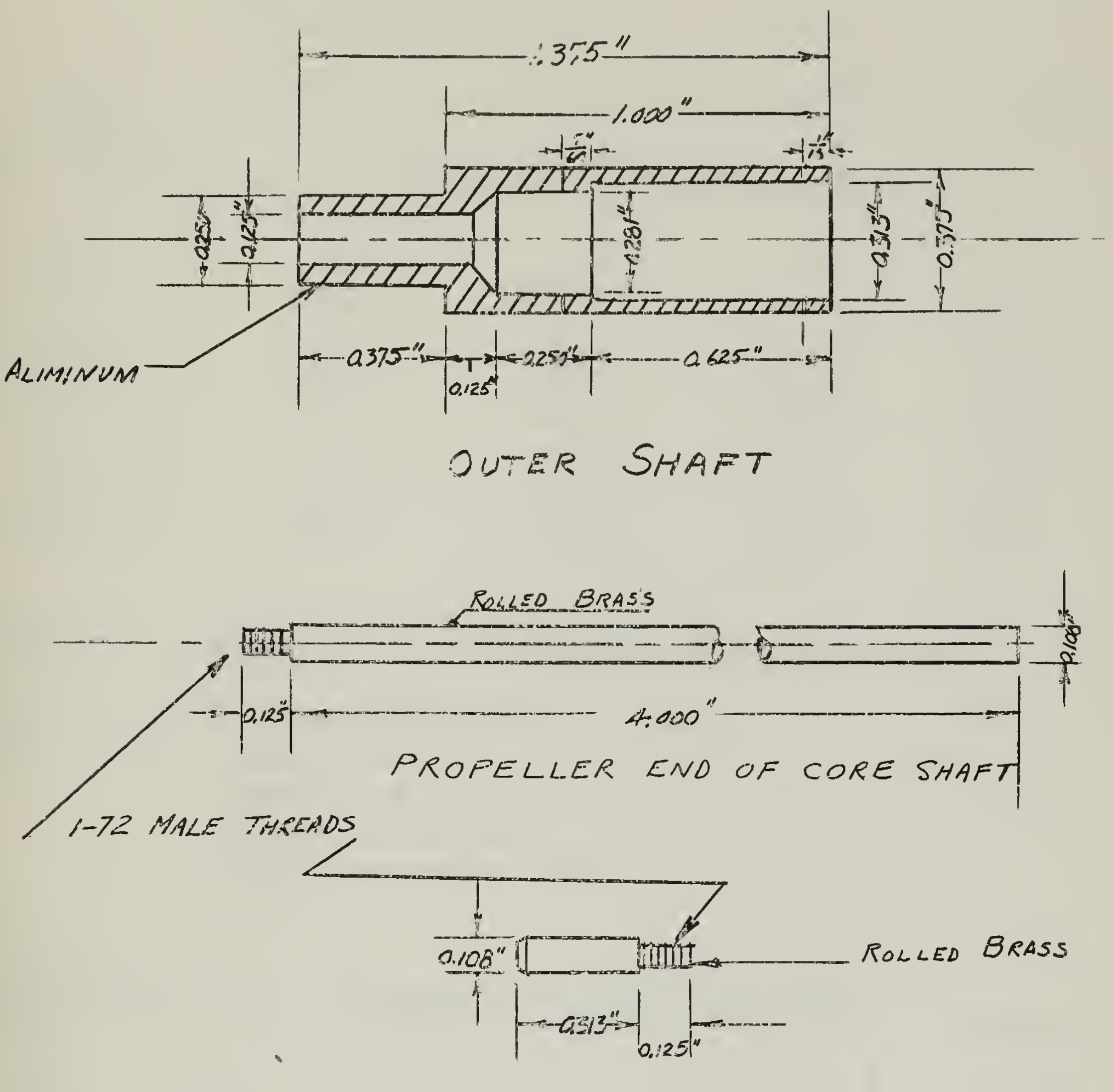
A. Slip ring noise observed using thin ring thrust gage. Recorded with Sanborn Single Channel Strain Gage Preamplifier and Recorder, Model 140. Gage Factor setting 1.72, Maximum Sensitivity, and Attenuator set at X20. This shows the noise before the rings were polished.



B. Slip-ring noise and deflections due to load pulses of approximately one and one-half pounds after slip-rings had been sanded with 6/0 sandpaper and polished with canvas. All settings on recorder as described above.

FIGURE XXIII

DETAILS OF LVDT THRUST GAGE SHAFT



FORWARD END OF CORE SHAFT

BLS
JAB
5/1/56

SCALE: TWICE FULL SIZE

APPENDIX B. Tabulated Data

TABLE I - Results of Lucite Torque Gage Static Calibration

Recorder Used: Sanborn Model 150 Dual Channel
Dial Settings: Gage Factor 2.0
Attenuation X2
Maximum Sensitivity
Zero Suppression "Out"

Applied Torque (in. - ounces)	Run #1	Run#2	Run#3	Run#4
	Millimeters Stylus Deflection			
0.308	6.0	5.0	6.1	5.5
0.616	12.5	11.7	12.6	11.5
0.924	18.7	18.0	19.3	17.5
1.232	25.0	25.0	25.0	24.0
1.386	28.7	27.5	27.0	27.5
1.540	32.2	30.0	31.8	31.0
1.694	35.5	32.0	34.5	33.3
1.848	38.7	37.0	37.5	36.5

TABLE II - Results of Lucite Torque Gage Static Calibration

Recorder Used: Sanborn Model 150 Dual Channel
Dial Settings: Gage Factor 2.0
Attenuation X5
Maximum Sensitivity
Zero Suppression "Out"

Applied Torque (in-ounces)	Millimeters of Stylus Deflection					
0.77	7.0	6.0	7.0	6.0	6.5	5.5
1.54	12.7	12.3	13.5	12.0	12.0	12.0
2.31	19.0	18.6	20.0	18.0	18.5	18.0
3.08	25.5	25.5	26.0	24.0	24.0	24.0
3.85	31.5	---	32.0	30.0	29.5	30.0

TABLE III - Results of Thin Ring Thrust Gage Calibration

Recorder Used: Sanborn Model 140 Single Channel
Dial Settings: Gage Factor 1.72
Attenuation X1
Maximum Sensitivity
Zero Suppression "Out"

Thrust Weight (Pounds)	Millimeters of Stylus Deflection
0.022	4.00
0.044	7.75
0.088	15.50
0.132	23.25
0.176	30.75

TABLE V - Results of Static Calibration of Differential Transformer Thrust Gage. Increments above a base load of 0.110 pounds

Recorder Used: Sanborn Model 150 Dual Channel
Dial Settings: Gage Factor 10.0
Attenuation X20
Sensitivity adjusted to obtain approximately 10 millimeters of stylus deflection for 0.044 pound load increment
Zero Suppression "Out"

Thrust Applied (Pounds) (Increments above 0.11#)	Run # 1	Run#2
	Millimeters of Stylus Deflection	
0.044	10.0	9.5
0.088	20.0	19.4
0.132	27.7	28.9
0.176	36.4	36.8

Note: Recorder Balance readjusted prior to each run

TABLE IV - Results of Static Calibration of Differential Transformer Thrust Gage

Applied Thrust (Pounds)	Run#1	Run#2	Run#3	Run#4	Run#5	Run#6	Run#7	Run#8	Run#9	Run#10
0.044	8.0	6.6	6.7	8.0	7.5	7.4	7.5	7.6	7.8	6.8
0.088	13.3	13.5	12.9	13.0	13.1	12.3	14.0	14.1	13.6	12.6
0.132	18.5	17.7	17.7	18.7	18.1	17.6	19.0	19.2	20.0	17.6
0.176	23.2	22.5	22.9	23.1	23.5	23.1	23.4	23.4	25.0	23.0
0.220	27.4	27.4	27.6	28.0	27.6	28.2	28.5	27.7	29.0	27.6
0.330	36.9	37.2	37.4	36.9	37.9	36.9	37.0	37.1	38.2	37.1
0.440	44.4	44.8	45.4	44.2	44.8	44.5	44.1	44.1	44.4	44.6

Recorder Used: Sanborn Model 150 Dual Channel

Dial Settings
Gage Factor 10.0
Attenuation X50
Sensitivity adjusted to obtain approximately 44 millimeters of stylus deflection for maximum load of 0.44 pound.
Zero Suppression "Out"

Notes:
1. All loads were dithered
2. Recorder was balanced before Run #1 and checked before Run#6

APPENDIX C SUPPLEMENTARY CALCULATIONS

1. Calculation of Series 60 Block 0.60 Model Torque and Thrust

a. Notation

J = Speed coefficient

V = Ship speed in feet per second

N = Ship propeller revolutions per second

D = Ship propeller diameter in feet

T_s = Ship thrust in pounds

Q_s = Ship torque in pounds - feet

ρ = Density of fresh water = 1.938 slugs

K_t = Thrust coefficient

K_q = Torque coefficient

T_m = Model thrust in pounds

Q_m = Model torque in pounds-feet

L_s = Ship length in feet

L_m = Model length in feet

λ = Ratio of L_s to L_m

b. Formulae

$$J = \frac{V}{ND} \tag{1}$$

$$T_s = K_t \rho D^4 N^2 \tag{2}$$

$$Q_s = Kq \rho D^5 N^2 \tag{3}$$

$$T_m = \frac{T_s}{\lambda^3} \tag{4}$$

$$Q_m = \frac{Q_s}{\lambda^4} \tag{5}$$

c. Calculation of Model Thrust (Figures referred to are in Ref. 18)

i. Assume: V_s = 25 knots = 42,25 ft/sec

L_s = 600 ft.

ii. From Figure 4(c) obtain N = 146 rpm = 2,435 rps

iii. Using propeller number 3378, D = 22.40 feet

iv. $J = \dfrac{42.25}{2.435 \times 22.40} = 0.775$

v. From Figure 4(b) $K_t = 0,186$

$K_Q = 0.033$

vi. From formula (2)

$$T_s = (0.186)(1.938)(22.40)^4(2.435)^2$$

$$= 5.38 \times 10^5 \text{ pounds}$$

vii. From formula (4)

$$T_m = \frac{5.38 \times 10^5}{(120)^3} = 0.31 \text{ pounds}$$

This is considered to be the maximum mean thrust at which the model will be run.

d. Calculation of Model Torque

i. From formula (3) $Q_s = (0.033)(1.938)(22.40)^5(2.435)^2$

$= 2.135 \times 10^{+6}$ pounds feet

ii. From formula (5)

$$Q_m = \frac{2.135 \times 10^6}{(120)^4} = \frac{2.135 \times 10^6}{2.076 \times 10^8}$$

= 0.01028 pounds feet; or converting to inch-ounces

iii. Q_m = 12 x 16 x 0.01028 = 1.97 inch-ounces

which is considered to be the greatest mean torque at which the model would be operated.

2. Calculation of Strain Developed in Saran Thrust Gage

For details of the procedure used to calibrate a recorder to read strains, see Reference (26) pp. 3-4 and 3-5.

[illegible]

a. Calibration deflection = $\dfrac{\text{bridge resistance}}{\text{(number active gages) (GF)(basic sensitivity)}}$

b. The number of active gages is two, the gage factor (individual is 1.72, and the basic sensitivity chosen was 20 micro-inches per inch per centimeter. The calibration deflection then is calculated to be

$$C.D. = \frac{120}{(2)(1.72)(20)} = 17.4 \text{ millimeters}$$

c. With the calibration set for this value, a 0.25 pound load (X1 attenuation) was found to produce a stylus deflection of 18 millimeters.

d. From the relation

$$\text{strain} = (\text{cm. deflection})(\text{basic sensitivity})(\text{attenuation})$$

we obtain

$$e = (1.8)\,(20)(1) = 36 \text{ micro-inches per inch}$$

3. Calculation of Gage Factor of Lucite Torque Shaft.

Details of method are given in Ref. (25). The procedure is as follows:

a. Gage Factor is set to some convenient value. Illustrating with Data of Run#7 from Table II of Appendix B, the G.F. = 2.0

b. After normal balancing procedure,weights were added to produce torques thereby causing stylus deflections. In this example 3.85 inch-ounces of torque produced 32.0 millimeters of stylus deflection.

c. The Zero Suppression switch is turned to "In" and Zero Suppression dial advanced from zero until the stylus deflection has been reduced to zero. The readings of the Zero Suppression dial at this time was noted to be 102.

d. Now, the Gage Factor is the output voltage per input voltage per unit load applied.

d. Now, the Gage Factor is the output voltage per input voltage per unit load applied.

Using the reading previously noted, this can be calculated following the regular procedure in the following manner:

$$G.F. = \frac{(G.F.\ Setting)}{(applied\ load)} \times \frac{Zero\ Suppres.\ Dial\ Reading}{1000}$$

$$G.F. = \frac{2\ mv/v}{3.85\ in\text{-}oz.} \left(\frac{102}{1000}\right)$$

G.F. = 53 microvolts output/volt input/inch-ounce torque

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